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MOTOR VEHICLE ENGINEERING

ENGINES

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MOTOR VEHICLE ENGINEERING

ENGINES

(FOR AUTOMOBILES, TRUCKS AND TRACTORS)

BY

ETHELBERT FAVARY

MEMBER SOCIETY OF AUTOMOTIVE ENGINEERS; CONSULTING ENGINEER; LECTURER ON MOTOR VEHICLE DESIGN, COOPER UNION

SECOND EDITION

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PREFACE TO THE SECOND EDITION

The necessity of revising certain portions of this book having arisen, the author took advantage of the opportunity to make a more thorough revision of the entire work than was originally contemplated. As a result many chapters have been entirely rewritten, numerous formulæ changed to show their derivation from standard practice, and practical data added or brought up to date.

The author requests that draftsmen, designers and others using this book advise him of any additional data relating to the design and testing of engines, which they feel should be added to future editions of the book. Every effort will be made to meet these suggestions in subsequent revisions or in the next volume of this series, "Chassis Design," which is now in preparation.

ETHELBERT FAVARY.

NEW YORK CITY,
February, 1920.

416882

PREFACE TO FIRST EDITION

The object of this book is to give in a concise manner and in simple language, the information needed by the designer and the automobile engineer in their everyday work, and with only the simplest of mathematics.

The book is also intended for draftsmen, technical graduates, mechanics and others, engaged in different branches of engineering, who wish to obtain a technical training in motor vehicle engineering, including design and testing.

Throughout this volume higher mathematics have been eliminated as the author has found from practical experience that a majority of those seeking information on engineering subjects, are conversant with elementary algebra only and perhaps some trigonometry; the book can therefore be readily followed by those whose mathematical knowledge is limited or has become rusty.

The author endeavors to bring the information up-to-date, and includes data of the latest designs, specially gathered for this purpose from the leading manufacturers. Truck engine design is treated exhaustively on account of the ever-increasing importance of this subject.

In short, the book illustrates modern motor vehicle engineering practice and the formulæ for determining the strength and dimensions of the various parts forming the engine of the present day. Examples of standard designs of all parts of engines are included to make it a thorough handbook on the subject.

The author wishes to express his great appreciation to the following Motor Manufacturers for furnishing material, which enabled him to make the data truly representative American practice:

The Pierce Arrow Motor Co.
Continental Motor Corp.
Packard Motor Co.
International Motor Co.
The Buda Company
The Hercules Motor Mfg. Co.
Haynes Automobile Co.
Olds Motor Works
Waukesha Motor Co.
Briscoe Motor Corp.
Falls Motors Corp.
Ford Motor Co.
Gray Motor Co.
Hudson Motor Car Co.
Hall Scott Motor Co., Inc.
The Kelly-Springfield Motor Truck Co.
The Locomobile Co. of America
Lycoming Foundry & Machine Co.
Moline Plow Co.

Reo Motor Car Co.
The White Co.
The National Motor Car & Vehicle Corp.
H. H. Franklin Mfg. Co.
King Motor Car Co.
Cadillac Motor Co.
The Peerless Motor Car Co.
Wisconsin Motor Mfg. Co.
The Nash Motors Co.
North American Motors Co.
Premier Motor Car Co.
The Studebaker Corp.
Willys-Overland Inc.
Apperson Bros. Automobile Co.
Chevrolet Motor Co.
Dodge Bros.

The readiness with which many manufacturers of Motor Vehicles and engines, especially the large concerns, supplied detail drawings, is significant of a new epoch, in contradistinction to the secrecy with which such details were formerly guarded. Manufacturers now realize that the dissemination of this science of *automotive engineering* is not only of benefit to the country at large, but is directly and indirectly of great benefit to themselves by placing at their disposal a larger number of highly trained engineers, designers, and draftsmen, and it will also help to familiarize large users of trucks and automobiles with the technical details of the motors described.

The author is also indebted to Mr. Heldt's "The Gasoline Automobile" which he frequently consulted; to Mr. John Younger for drawings of the Class B, U. S. A. Military Truck Engine, and to Mr. C. F. Scott, of the Sprague Electric Works, for preparing the description of the electric dynamometer.

He also takes this opportunity to express his great appreciation to the personnel of the Society of Automotive Engineers, who at all times showed their readiness to supply data or information whenever requested.

The author would be grateful for any suggestions toward improving subsequent editions of the book, and for drawing his attention to any errors which usually are unavoidable in a first issue.

This volume on engines is the first of a series; a second volume treating of the chassis in general, and a third on carburetion of liquid fuels of all kinds, are contemplated by the author.

If these works contribute to the better training of even a small number of men engaged in this large industry, the author will feel that his purpose has been accomplished.

ETHELBERT FAVARY.

NEW YORK CITY,
January, 1919.

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CHAPTER I

ENGINE PRINCIPLES

The internal combustion motor, often called the hydro-carbon engine, using gasoline, kerosene or alcohol as fuel, is almost universally employed in motor vehicles to-day. There are two distinct types of internal combustion motors which have been applied to automobiles, the two-cycle and the four-cycle, but the four-cycle engine is the one in general use.

The four-cycle is often called the "Otto Cycle" as Dr. Otto was the first to use it in practice, although it was first proposed by Beau de Rochas. This cycle requires four strokes of the piston for its completion, hence the name four-cycle, which in reality should be four-part cycle or four-stroke cycle. The following is the cycle of operation:

First Stroke.—On the first or outward stroke of the piston (that is away from the piston head) a mixture of gasoline vapor and air is drawn into the cylinder through the inlet valve *A* (see Fig. 1) similarly as a pump is filled with water.

Second Stroke.—The piston returns and before the mixture of air and vapor escapes, the inlet valve is closed. The mixture is therefore trapped in the cylinder and as the piston is completing the second stroke, the gas mixture (the explosive charge), is compressed to from 60 to 90 lb. per square inch above atmospheric pressure. Just before the second stroke (usually called the compression stroke) is completed, ignition takes place, that is to say, an electric spark is produced inside the cylinder, igniting the explosive mixture. As the mixture is ignited an explosion occurs, the pressure of which rises to about four times that of the compression pressure. An explosion is simply an extremely rapid rise in temperature; the high temperature tends to greatly

expand the gas, but being confined in the cylinder it will result in great pressure.

Third Stroke.—On the third stroke of the piston the ignited gas expands and transmits the power, derived from the heat of combustion, to the piston, for the piston is the only portion which can give way. From the piston the power is transmitted through a connecting-rod to the crankshaft, and thence to the driving wheels of the automobile. Just before the power stroke, *i.e.*, the third stroke, is completed, when the piston is about 1/5 from its final point of travel, the exhaust valve is opened and the

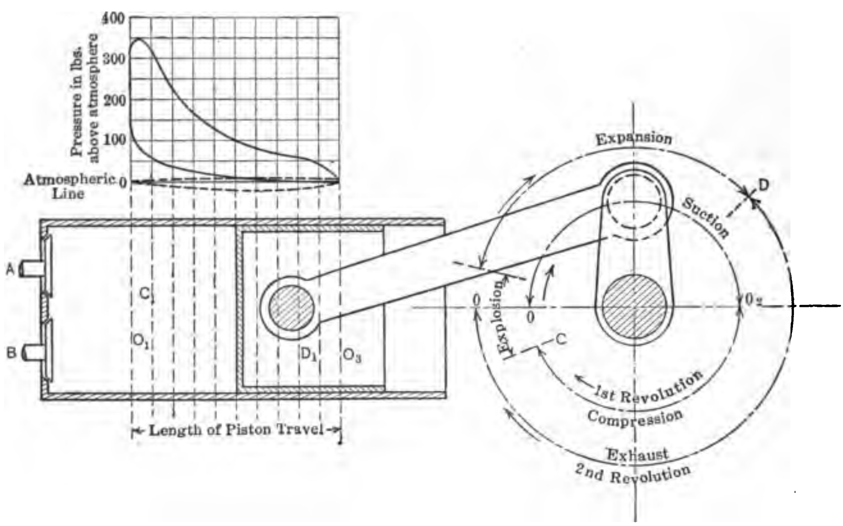


FIG. 1.—Complete cycle of operation of four-cycle engine.

burned gas commences to escape through this valve to the atmosphere.

Fourth Stroke.—The exhaust valve is open and the piston clears the cylinder of all or nearly all the burned gas, and when the piston completes this stroke (when it is at the inner dead center), or a trifle later, the exhaust valve is closed.

The next stroke is another suction stroke and the same operation is repeated as before.

Internal combustion engines applied to automobiles are single acting, that is, only one end of the cylinder is used, there being no crosshead as in steam engines, and one end of the cylinder is open.

Figure 1 shows a diagrammatic representation of the four-cycle engine. Above the cylinder is shown what is known as an indicator diagram, which indicates what is taking place within the cylinder during the cycle of operation.

The piston stroke, *i.e.*, the length of piston travel, is divided into ten equal parts. The vertical numbers on the diagram give the pressure in pounds per square inch above the atmospheric pressure. By the aid of this diagram we can annalize what is taking place inside the cylinder. On the first stroke of the piston (from O to O_2 on the crank circle) there is a slight vacuum, as can be seen from the dotted curve slightly below the zero line on the diagram. During this stroke, suction takes place, that is to say, the piston draws into the cylinder the explosive mixture through the inlet valve A. When the piston completes the first stroke, which is the first half of the first revolution of the crankshaft, the inlet valve is closed. On the second half of the first revolution, that is, on the return stroke of the piston, the valves being closed, the mixture in the cylinder is compressed. The reason for compressing this mixture is to obtain a higher pressure from the explosion; it has been found from experiments, that the higher the compression of the explosive mixture before ignition, the higher is the resultant pressure on explosion. With gasoline vapor, the resultant pressure varies from three to five times the value of compression. When the piston reaches C_1, or the point C on the crankshaft circle, ignition occurs. As seen, ignition takes place before the end of the second stroke is reached; the reason being, that the total gas mixture does not ignite instantaneously but gradually, and as the greatest pressure derived from the explosion should occur at the beginning of the third stroke (in order to give the gas an opportunity to expand as much as possible), it is necessary to ignite the charge before the end of the second stroke is reached. Another reason for igniting the mixture before the end of the compression stroke, is to have the ignition of the greatest quantity of the explosive mixture occur at a time when the compression is highest. The maximum pressure resulting from the explosion, is in this case, as shown in the diagram, above 300 lb.

When the third stroke is about 1/5 from the outer dead center, shown at D on the crankshaft circle and at D_1 on the cylinder, that is, when the pressure in the cylinder drops to about 50 lb. per square inch, the exhaust valve B is opened, to permit the

escape of the burned gas. The exhaust valve remains open during the whole time of the fourth stroke, when the piston expels the burned gas. On this stroke there is a small back pressure as shown by the dotted curve above the zero line of the diagram. The reason for opening the exhaust valve before the end of the third stroke, is to avoid the high back pressure, which would necessarily ensue, if the valve were opened later. By opening the exhaust valve a little earlier, the pressure in the cylinder will drop to nearly atmospheric pressure before the piston starts its return stroke, and a longer time is given for the escape of the burned gas. On completion of this stroke, the exhaust valve closes and the piston is ready to repeat the entire operation.

In high-speed engines, it is customary to close the inlet valve, not at the end of the first stroke, but after the piston has started its second stroke, in order to get a greater volume of mixture or a greater amount of gas into the cylinder. On the suction stroke, the incoming gas mixture has a certain amount of inertia, just the same as the motion of a column of water; on finishing the suction stroke, this inertia of the gas mixture will cause it to enter the cylinder for some time after the end of the stroke is reached.

There is still another engine manufactured, which might be mentioned under the heading of the four-cycle type, as an explosion occurs at every fourth stroke of the piston. This Engine is called the Diesel engine, according to the name of the inventor. Although this engine has not been applied to automobiles, a brief explanation of its principle may not be out of order.

In the Diesel engine, the first or outward stroke of the piston draws into the cylinder a charge of pure air. On the second stroke this charge of fresh air is compressed to about 500 to 600 lb. per square inch. The temperature at the end of compression is naturally very high. The fuel is introduced at the beginning of the third stroke, under a pressure higher than that of compression, and takes fire immediately, due to the high temperature inside the cylinder. The fuel, whether oil, or gasoline, forced into the cylinder when the piston commences the third stroke, is only cut off after it has traveled about 10 per cent. of its total stroke. The fuel is burning as it enters, due to the high temperature in the cylinder, and therefore, the high pressure is maintained; when the supply is cut off,

the product of combustion expands. On the next stroke, the burned gas is expelled from the cylinder. As in this engine the compression is very high, the clearance space must naturally be very small, therefore, on the fourth stroke, nearly all the burned gas is expelled (more than in the ordinary four-cycle type, with its larger clearance space). The importance of such a motor cannot be over-estimated, as any low grade petroleum oil can be used for fuel. So far this engine gives the highest fuel efficiency of all internal combustion engines.

The two-cycle engine is to-day very rarely used in motor vehicles, and for this reason it is not specifically discussed in this volume.

CHAPTER II

COMPRESSION, COMBUSTION AND EXPANSION OF GAS

In this chapter the element that gives the power in internal combustion engines will be considered. A knowledge of the subject is indispensable to the designer, as will be realized in subsequent chapters treating of practical designs of engines.

Suppose we take a cylinder and piston, fill the cylinder with gas (air, or vapors of gasoline, alcohol or petroleum oil are all termed "gas"), and assume that there is absolutely no leakage through the cylinder; if the gas inside the cylinder is compressed, its volume is decreased, but there will be a pressure in the cylinder. When compressing a gas, its temperature is raised at the same time, on account of the energy which has to be expended in the compression.

If the piston is moved to and fro, the gas in the cylinder will alternately be compressed or expanded, and IF WE SUPPOSE that the temperature of the gas be kept constant, we find, if an indicator, or a pressure gauge, is attached to the cylinder, that the pressure inside varies in direct proportions as the volume is decreased. For instance, assuming the length of the cylinder space between the cylinder head and the piston to be 10 inches, if the piston is moved halfway in, the pressure in the cylinder will be double the initial pressure. If the initial pressure is 14.7 lb. per square inch, which it would be in this case, the compression would be 29.4 lb. per square inch. (The atmospheric pressure is 14.7 lb per square inch = 1 atmosphere.) If a curve is traced of the values for the different proportions of volume and pressure so obtained, it is called the Isothermal curve. As seen from the diagram Fig. 2, if the volume, or cylinder length, be halved, the pressure is doubled (2×14.7 is 29.4 lb.). At $\frac{1}{4}$ volume, the pressure is $4 \times 14.7 = 58.8$, and so on for the different proportions. Boyle was the first to formulate this law and it is stated in the following: If the temperature of a gas is kept constant, its pressure will vary inversely as the volume it

occupies. It is represented by the formula $P \times V = C$, that is Pressure $\times$ Volume = Constant. From this we have $\frac{C}{P} = V$, and $\frac{C}{V} = P$.

In practice the temperature cannot be kept constant, therefore a third factor, that is the heat developed during compression, must be considered. The volume of gas expands when its temperature is raised and contracts by the loss of heat. The following was first formulated by Gay Lussac: Given a certain amount of gas in a cylinder with a free piston (that is a piston without resistance) the volume of the gas will give a proportional

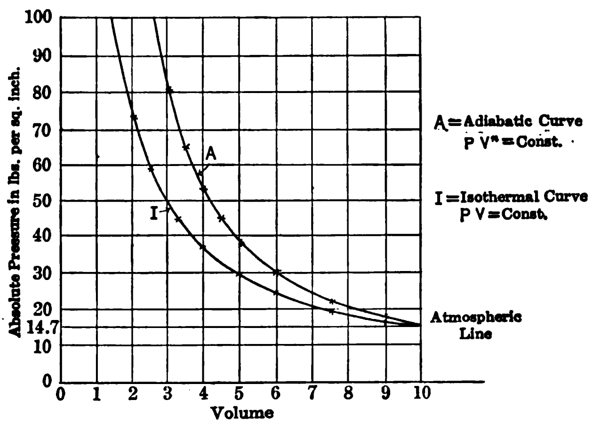


FIG. 2.—Expansion of gas.

displacement of the free piston, equal to 1/492 part of the total volume of the cylinder for each degree Fahrenheit. And therefore: When a fixed volume of gas is increased or decreased by an increase or decrease of heat, it varies in the same proportion of 1/492 part of its pressure for each degree Fahrenheit change in temperature.

The value of 492 (exact value is 491.65), represents the absolute zero cold below the freezing point of water. In the Fahrenheit scale, the absolute zero of cold would be $-491.65°$ or $491.65° - 32° = 459.65°$F. below zero Fahrenheit. According to Boyle's Law there are only two characteristics of a gas to be considered, that is volume and pressure; according to the law of Gay Lussac, which is usually called the second law of gases, a third char-

acteristic is added, viz., the temperature at which the operation takes place. This law is called the Adiabatic Law.

If we take the cylinder and piston before mentioned, and move the piston rapidly to and fro, and if an indicator were attached to the cylinder, it would show a curve corresponding not with Boyle's Law or the Isothermal Law, but with the Adiabatic Law, and the curve which takes into account temperature changes as well, is called the Adiabatic curve of a gas. The expansion of the gas $\frac{1}{492}$ of its volume for every degree Fahrenheit, from 32°, added to its temperature, is equal to the decimal .0020325, and is the coefficient of expansion from 32° for each degree rise in temperature of the Fahrenheit scale.

Let us work out an example:

For instance, say we wish to find the pressure of a certain volume of air or gas if the temperature is raised from 70°F. to 2000°F. and if the volume is kept constant. The temperature increase will be 1930°F. The ratio of expansion will be $\frac{1}{(492 - 32) + 70} = \frac{1}{530} = .0018868.$

For any given volume of gas its gauge pressure, at constant volume, can be found as follows:

Ratio $\times$ acquired temperature $\times$ initial pressure = gauge pressure.

It should be remembered that this formula is not absolutely true as there is a small difference due to the expansion of a dry gas, the moisture in the atmosphere and the vapor of water formed in the combustion chamber of explosive engines. For practical purposes however, it is sufficiently accurate to use this formula:

Working out our example we have:

$$.0018868 \times 1930 \times 14.7 = 53.53 \text{ lb. (gauge pressure)}$$

Another convenient formula for getting the absolute pressure is

$$\frac{\text{Atmospheric pressure} \times (\text{absolute temperature} + \text{final temperature})}{\text{Absolute temperature} + \text{initial temperature}}$$

Using the same formula as before, we have:

$$\frac{14.7 \times (460° + 2000)}{460 + 70} = 68.23 \text{ lb. (absolute pressure)}$$

and $68.23 - 14.7 = 53.53$ lb. gauge pressure.

For obtaining the expansion, expressed in volumes, for a certain given increase in temperature, we have the following formula:

$$\frac{\text{Volume} \times (\text{absolute temperature} + \text{final temperature})}{\text{Absolute temperature} + \text{initial temperature}} = \left\{ \begin{array}{l} \text{Heated} \\ \text{volume.} \end{array} \right.$$

Working out the example we had before:

$$\frac{1 \times (460^\circ + 2000^\circ)}{460^\circ + 70^\circ} = \frac{2460}{530} = 4.6415 \text{ volumes,}$$

that is to say if we heat a given volume of gas from 70°F. to 2000°F., its volume will be 4.6415 times as large (as it was before heating) keeping the pressure the same; in this case, the atmospheric pressure.

If we raised the temperature of the gas in a confined space, its pressure due to expansion, will be, in the above example: $4.6415 \times 14.7 = 68.23$ lb. absolute, or $68.23 - 14.7 = 53.53$ lb. gauge pressure.

If we wish to find the acquired heat we have the formula:

$$\frac{\text{Absolute pressure} \times (\text{absolute temperature} + \text{initial temperature})}{\text{Initial absolute pressure}}$$

$= \text{Absolute temperature} + \text{temperature at combustion}$, and by subtracting the absolute temperature we get the acquired temperature. By using the example we had before, we have:

$$\frac{68.23 \times (460^\circ + 70^\circ)}{14.7} = 2460^\circ$$

and $2460^\circ - 460^\circ = 2000^\circ$, which is the theoretical heat of combustion, in this case.

By looking at the curves, Fig. 2, it is possible to see at a glance the pressure in a cylinder for every position of the piston during compression. The "*I*" curve shows the pressure if it were possible to keep the temperature constant; in practice this cannot be done however. The "*A*" curve shows the actual pressure including that due to the rise in temperature.

In drawing the curves we proceed as follows: Boyle's Law, as we have seen, is $PV = C$, where P is the pressure, V the volume and C a constant quantity. If P and V are the pressure and volume before compressing the gas, and P_2 and V_2 the pressure and volume after compression, we have $P_1V_1 = C$; also $P_2V_2 = C$, therefore $P_1V_1 = P_2V_2$, hence $P_2 = \frac{P_1V_1}{V_2}$.

In this case $P_1 = 14.7$, the atmospheric pressure, and V_1 the entire volume before compression which, here, is 10.

$$\text{If } V_2 = 1,\ P_2 = \frac{10 \times 14.7}{1} = 147.0 \text{ lbs}$$

$$\text{If } V_2 = 2,\ P_2 = \frac{10 \times 14.7}{2} = 73.5 \text{ ''}$$

$$\text{If } V_2 = 2.5,\ P_2 = \frac{10 \times 14.7}{2.5} = 58.8 \text{ ''}$$

$$\text{If } V_2 = 3.33,\ P_2 = \frac{10 \times 14.7}{3.33} = 44.1 \text{ ''}$$

$$\text{If } V_2 = 4,\ P_2 = \frac{10 \times 14.7}{4} = 36.75 \text{ ''}$$

$$\text{If } V_2 = 5,\ P_2 = \frac{10 \times 14.7}{5} = 29.4 \text{ ''}$$

$$\text{If } V_2 = 6,\ P_2 = \frac{10 \times 14.7}{6} = 24.5 \text{ ''}$$

$$\text{If } V_2 = 7.5,\ P_2 = \frac{10 \times 14.7}{7.5} = 19.6 \text{ ''}$$

$$\text{If } V_2 = 9,\ P_2 = \frac{10 \times 14.7}{9} = 16.33 \text{ ''}$$

$$\text{If } V_2 = 10,\ P_2 = \frac{10 \times 14.7}{10} = 14.7 \text{ ''}$$

The values of P_2 so found are then marked on squared paper by small crosses, as shown, for the different values V_2, afterward drawing a curve through the points marked by the crosses.

For drawing the adiabatic curve we have the Adiabatic Law

$$PV^n = C; \text{ then } P_1V_1^n = P_2V_2^n, \text{ and } P_2 = \frac{P_1V_1^n}{V_2^n} = P_1\left(\frac{V_1}{V_2}\right)^n.$$

To solve this equation it is necessary to use logarithmic tables.

The next chapter contains a table giving all the figures required in practice without having to resort to logarithms.

For a perfect gas, like air, the exponent $n = 1.41$.

Let us work out an example when V_2 is 4, and V_1 and P_1 the same as in the last example. We want to find P_2 which $= P_1\left(\frac{V_1}{V_2}\right)^n = 14.7\left(\frac{10}{4}\right)^{1.41}$. Log. of $\frac{10}{4}$ *i.e.*, logarithm of 2.5 = .3979; .3979 × 1.41 = .56104. Antilog of .56104 = 3.638; 14.7 × 3.638 = 53.5.

In this manner the values for the different volumes were figured.

If $V_2 = 2$, $P_2 = 146.2$
If $V_2 = 3$, $P_2 = 81.1$
If $V_2 = 3.5$, $P_2 = 64.5$
If $V_2 = 4$, $P_2 = 53.5$
If $V_2 = 4.5$, $P_2 = 45.3$
If $V_2 = 5$, $P_2 = 37.8$
If $V_2 = 6$, $P_2 = 30.1$
If $V_2 = 7.5$, $P_2 = 22.0$
If $V_2 = 9$, $P_2 = 17.4$
If $V_2 = 10$, $P_2 = 14.7$

CHAPTER III

CYLINDER CLEARANCE AND COMPRESSION.

In designing a gasoline motor one of the first considerations is the compression pressure and the percentage of clearance. From Chapter II it was seen that when compressing a gas, the pressure will vary approximately, according to the Adiabatic Law, which is PV^n = Constant, where P is the pressure, V the volume, and n, the exponent, is a value derived from practice; it is the ratio of the specific heat at constant pressure to the specific heat at constant volume. It has been found by practical tests that the exponent n varies as the initial pressure is increased or decreased. The official publication of the S. A. E. gives the following average figures as a result of a large number of tests:

if the initial pressure is 11½ lb. (absolute), $n = 1.21$;
if the initial pressure is 13 lb. (absolute), $n = 1.29$;
if the initial pressure is 14 lb. (absolute), $n = 1.34$.

The initial pressure in an engine will vary according to the speed, for the higher the speed (everything else being the same), the lower the initial pressure. This is caused chiefly by the increased friction of the gas in passing through the inlet pipe and valve passages.

From the above figures the pressure at the end of the compression stroke may be determined. For medium compression pressures the initial pressure is 13 lb. absolute and as we have seen above $n = 1.29$.

If P_1 and V_1 are the pressure and volume at the commencement of the stroke (that is, at the outer dead center of the piston), and P_2 and V_2 the pressure and volume at the end of the compression stroke (which is at the inner dead center), as was shown before, then $P_2 = P_1\left(\frac{V_1}{V_2}\right)^n$. Looking at Fig. 3, if the length of the piston stroke is shown by the dotted lines and full lines of the piston respectively, the ratio of compression would be $\frac{V_1}{V_2}$, and the clearance (expressed in Piston displacement) would be $\frac{V_2}{V_1 - V_2}$.

In other words, Clearance = Volume of Clearance divided by Volume of piston displacement. The ratio of compression is the total volume of the cylinder (including the clearance volume), divided by the volume of the clearance. Working out an example taking Fig. 3 for instance: (with the piston as shown at inner dead center), the clearance would be $\frac{2}{6-2} = .5$, or to have it in per cent., multiply by 100, which is 50 per cent. The ratio of compression would be $\frac{6}{2} = 3$.

For determining the pressure at the end of the compression stroke, we have the formula:

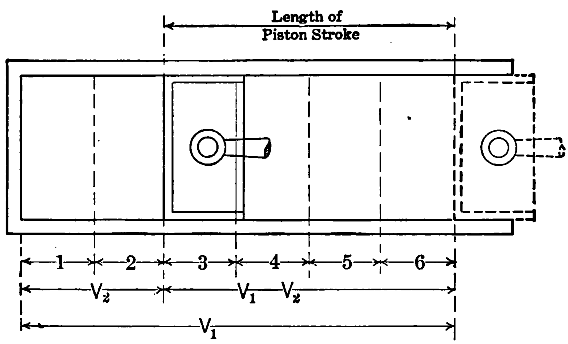


FIG. 3.

$P_2 = P_1\left(\frac{V_1}{V_2}\right)^n$. To work out this example it is necessary to use a table of logarithms. For those unfamiliar with logarithms, we are giving the figures determined with the various exponents n, as taken from the S. A. E. data sheets.

Column *A* is the ratio of compression.

Column *B* is the per cent. of clearance, corresponding to the ratio of compression, of column *A*.

Columns *C* gives the figures for low, medium and high compression pressures.

From the following table any compression pressure for a given ratio of compression, or a given clearance can be seen at a glance.

For instance: If it is required to find the compression pressure of a cylinder having a ratio of compression of 4.33 or a percentage of clearance 30, we can see from column *C*, that for an

initial pressure of 13 lb., the compression is 87 lb., absolute, and by subtracting 14.7 lb. (the atmospheric pressure) we have the gauge pressure 72.3.

TABLE I.—(FOR FINDING COMPRESSION PRESSURES IN GASOLINE ENGINES)

A Ratio of compression $\frac{V_1}{V_2}$	*B* Cylinder clearance in per cent. of piston displacement $\frac{V_2}{V_1 - V_2} \times 100$	*C* Compression pressure in lb. per square inch absolute		
		Initial pressure 11½ lb.	Initial pressure 13 lb.	Initial pressure 14 lb.
6.0	20	100	130	154
5.76	21	95	125	145
5.55	22	91	120	138
5.35	23	87	115	132
5.17	24	83	110	126
5.0	25	80	105	121
4.85	26	77	101	116
4.70	27	74	97	111
4.57	28	71½	94	107
4.45	29	69	90	103
4.33	30	67	87	99
4.23	31	65	84	96
4.125	32	63	81½	93
4.03	33	61½	79	91
3.94	34	60	77	88
3.86	35	58	75	85
3.78	36	56½	73	83
3.70	37	55	71	81
3.63	38	54	69	79
3.56	39	53	67½	77
3.5	40	52	66	75

In practice the initial pressure is not 14.7 (the atmospheric pressure) but on account of the resistance in the inlet passage, inlet pipe, etc., it is usually about 13 lb. therefore, the compression pressure of a cylinder having for instance, a ratio of compression of 5 is 105 lb. absolute (for an initial pressure of 13 lb.), and by deducting the atmospheric pressure we obtain a gauge pressure of 105 − 14.7 = 90.3 lb.

It must be remembered that the compression depends also upon the moisture in the atmosphere, the proportion of air in the explosive mixture, the fitting of the piston rings and valve

seats, etc. For practical purposes, however, the results in this table will usually be found satisfactory.

Gasoline vapor cannot be compressed in practice to more than about 90 lb. (gauge pressure) except under certain conditions. In a number of the latest designs the compression pressure averages about 80 lb. gauge pressure. In very high speed engines the compression may be carried as high as 110 lb. gauge pressure.

CHAPTER IV

THEORY OF FOUR-CYCLE ENGINE DISCLOSED BY INDICATOR DIAGRAM

Studying the indicator diagram is the best means of judging correct and faulty design of an engine. Keeping the various phases of this diagram in mind the student is enabled to design the motor for a desired performance, and to eliminate faults in design and construction.

The problem confronting the automobile engineer is to obtain the maximum efficiency and to reduce losses to a minimum.

In order to ascertain the efficiency of an engine it is first necessary to consider what is taking place in the cylinder during the cycle of operations.

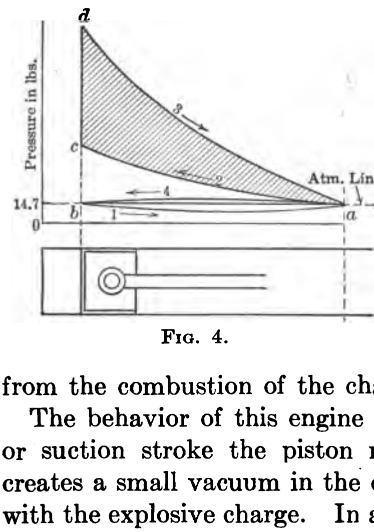


FIG. 4.

Let us consider an ideal four-cycle engine and assume there are no losses. An indicator diagram of such an engine is shown in Fig. 4.[1] In this case a–b corresponds to the length of the stroke, b–c to the compression pressure and c–d to the increase of pressure resulting from the combustion of the charge.

The behavior of this engine would be as follows: On the first or suction stroke the piston moving in an outward direction creates a small vacuum in the cylinder, in order to fill the latter with the explosive charge. In a perfect engine this suction curve should be as near the atmospheric line as possible.

The compression curve 2 should begin on the atmospheric line and should follow the Adiabatic Law PV^n = constant, where P is the pressure, V the volume, and n the ratio of the specific heat at constant pressure to the specific heat at constant volume. (This matter will be taken up in subsequent chapters.)

[1] These diagrams with data were collected by the author some years ago and were published by the *Horseless Age*, Oct. 2, 1907.

The curve *c–d* corresponds to the pressure arising from the explosion; in other words, to the rapid rise in temperature due to the ignition of the gaseous fuel. In an ideal engine this curve *c–d* should follow a straight line; in practice, however, as we shall see later, this is never the case. The expansion curve 3, like the compression curve 2, should follow the Adiabatic Law, and should end at atmospheric pressure, and the exhaust curve 4 should lie entirely on the atmospheric line. (The suction and the exhaust curve have been traced somewhat away from the atmospheric line, to bring them out more clearly.)

In this ideal engine there are no losses inside the cylinder, and the power available is determined by the area of the shaded section. It can be seen in this case that the expansion of the gas is carried to atmospheric pressure. In automobile engines this is impracticable on account of the comparatively short stroke, and further, on account of engine weight, which should be a minimum. In automobile work, therefore, we should not strive to obtain a diagram like that of Fig. 4, but one like Fig. 5, which is a diagram of an ideal automobile engine.

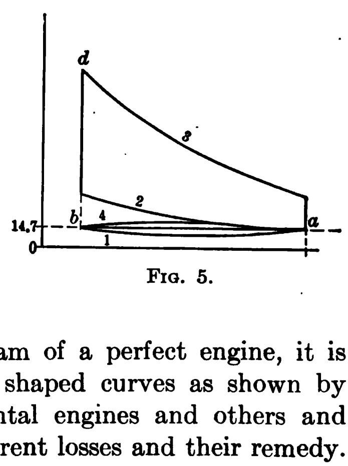


FIG. 5.

Having seen the indicator diagram of a perfect engine, it is possible now to consider different shaped curves as shown by diagrams obtained from experimental engines and others and to determine the causes of the different losses and their remedy.

The carburetor or fuel mixing device employed with automobile engines is usually at some distance from the inlet valve, thereby necessitating the employment of a pipe or tube for their interconnection. The office of the vacuum created by the suction stroke is to pump the charge from the carburetor through the inlet pipe, through the inlet valve and into the cylinder. In order to do this some power will be lost, corresponding in amount to that required to overcome:

1. The resistance in the carburetor itself.
2. The resistance of the inlet pipe.
3. The resistance of the inlet valve.

When the carburetor is working properly, and when the area of the inlet pipe and the inlet valve is correctly designed for the

maximum permissible speed of the gas, when sharp bends or elbows between the carburetor and the valve are eliminated, this loss will be very small, as shown by suction curves obtained from experimental engines. However, if these conditions are not fulfilled, the power so lost may be appreciable, and can be determined from the diagram, Fig. 6. In this diagram the suction line has been exaggerated, as in practice it is not so far below the atmospheric pressure. The shaded area below the atmospheric line gives the power lost due to the friction of the fresh charge. In this case, the portion *a–b* of the curve is horizontal, which shows that the speed of the gas through the valve passage was constant. In engines where the valve passage is too small the suction curve will decline toward *a*, as shown by the dotted line.

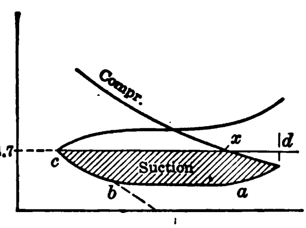


Fig. 6.

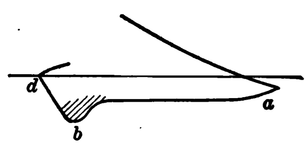


Fig. 7.

Some engines have their power considerably decreased by an improper timing or shaping of the cams, and the diagrams show these faults.

The inlet valve should not be opened until the pressure in the cylinder has fallen to atmospheric pressure, otherwise some of the exhaust gas will pass through the inlet valve into the inlet pipe. Late opening of the inlet valve can be recognized on the diagram by the line *d–b*, Fig. 7, where it is seen that the vacuum increases rapidly at the commencement of the stroke. At *b* the valve is opened, and the gas rushes first into the cylinder at a high speed, and gradually, as the vacuum decreases, the speed of the gas becomes normal. The total loss arising from a late opening is, in this case, very small. (It is larger, however, in high-speed engines and increases with the speed at which the gas passes through the valve opening.) It is represented by the shaded section in Fig. 7. Of more importance is the time of clo-

sure of the inlet valve. If this takes place too early, the vacuum will rapidly rise from the time of closure, as indicated by the line *b–c*, Fig. 8. In this instance the shaded section represents the power expended in creating the unnecessary vacuum in the cylinder. Of course, this loss is very small compared with the real harm done, arising from an incomplete charge, but of this we shall treat presently.

If the closure of the inlet valve takes place too late, a portion of the charge will be forced back into the inlet pipe and into the carburetor. This can easily be detected on the indicator diagram, Fig. 9. Here, at the end *b* of the suction stroke the vacuum will first drop to atmospheric pressure, *c*, and as the piston returns it will force the charge out of the cylinder at a slight pressure above atmosphere from *c* to *d*.

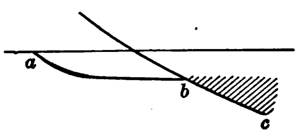


FIG. 8.

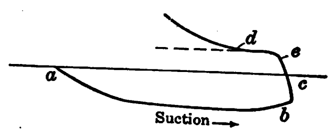


FIG. 9.

At *d* the valve closes when compression begins. If the closure occurred later still, the horizontal portion of the curve would simply be carried further along the dotted line, thereby decreasing the charge and consequently the compression. However, the inlet valve might be kept open for a short period after dead center to obtain a larger charge. The column of charge flowing through the inlet pipe into the cylinder has a certain inertia which will pump an additional portion of gas into the cylinder after the piston has started its return stroke. We see, then, that in timing the inlet valve it is important to have the closure just at the right moment, otherwise a loss will occur. In Fig. 9, if the inlet valve were closed at *e*, the volume of gas in the cylinder would be appreciably larger. When taking indicator diagrams from engines for the purpose of timing the valves it is important to do so at the speed at which the engine is mostly intended to be run, as the thermal efficiency will vary with the speed.

If in an automobile engine the beginning of the compression curve is a few pounds below atmospheric pressure, it results not only in a decreased compression pressure but also in a decreased quantity and an inferior quality of the charge, as the

cylinder will not be completely filled, and the proportion of burned gas to fresh mixture is larger.

The loss of charge can be determined by the point at which the compression curve crosses the atmospheric line. In Fig. 6, for instance, this point is marked x; the length of the diagram being the distance from c to d, the loss of the charge will be the distance dx divided by the distance dc, that is to say, $\frac{dx}{dc}$. If the distance dc is 10, and dx is 3, the loss of charge would be $\frac{3}{10}$ (3 divided by 10) which is .3 or 30 per cent.

COMPRESSION, FLAME PROPAGATION, ETC.

In a well-designed automobile engine the compression curve should begin around a pressure of 13 lb. absolute, and should follow the law $PV^n = C$, where n is about 1.29. Knowing this value, we can easily calculate the pressure for any particular point of the stroke or of the indicator diagram. (See Chapter III.)

In water-cooled automobile engines of to-day the compression pressure is about 80 lb. per square inch. If it were not for the high speed of these engines in many cases the compression could not be carried so far, owing to spontaneous ignition, but since the combustion does not take place instantaneously the piston passes the dead center before ignition of the total charge. In order to be able to obtain very high compressions the charge should be kept cool before entering the cylinder. There should be no sharp projections in the combustion chamber, and the valve seats should be properly water jacketed. In many engines on the market the fresh charge is purposely made to pass close to or is surrounded by the exhaust pipe in order to heat it for the purpose of a more thorough vaporization of the fuel. This is distinctly bad from the point of compression and volume of charge, but is necessary for the purpose named. It is a case of curing one evil with another but as the engine shows a higher fuel efficiency, it is made use of. On the other hand, a cooler charge will often result in more power from a given engine, even though the fuel efficiency is lower. With high-grade gasoline, heating of the fresh charge by the exhaust is not necessary, but the last few years, gasoline has become much poorer in quality, and unless heated, some of the charge will condense. It is well

known that the higher the compression pressure the higher is the efficiency of the engine, and the larger the volume of the charge in the cylinder the greater the power derived therefrom. In high compression engines the clearance is smaller, which means that the proportion of burned gas is smaller. The latter will assist in keeping the fresh charge cooler, thereby facilitating a larger and purer charge and a higher compression, or rather, preventing premature ignition to a considerable extent.

The effect of the compression pressure upon the thermal efficiency (which is the per cent. of the heat units of the fuel that is transformed into work) can be seen approximately by the following table:

Compression pressure in lb. per sq. in. above atmosphere	Thermal efficiency as computed from the indicator diagram, per cent.
30	28.3
37	31.2
44	33.0
50	34.5
59	36.8
67	39.0
75	41.5
81	42.0

High compression and pure charges have also a very important bearing upon the rapidity of combustion or the "flame propagation" of the charge (see Figs. 10 and 11).

The three diagrams *A*, *B* and *C*, Fig. 10, give curves showing different flame propagations due to different compressions, and in order to obtain as favorable curves as these the point of ignition had to be shifted; *a* gives the point of ignition and *b* the point where the total mixture is practically ignited and expansion begins. The different compression pressures when these curves were taken were approximately 20 lb., 40 lb. and 60 lb., respectively, and the engine speed was about 700 r.p.m. Figure 11 gives two curves, *a* and *b*, of the same engine and for the same compression and point of ignition, except that when curve *b* was taken there was a greater amount of burned gas in the cylinder. At first sight it would appear as if the compression were not the same in both cases. This, however, is not so, for after ignition took place at *c*, the curve *a*, corresponding to a purer charge, rises rapidly at the end of the stroke to *e*, whereas curve *b* rises in the same time only to *d*. If ignition

had been advanced on curve *b* its peak would be slightly higher, as the pressure at the end of the stroke would be higher, which means a higher temperature and therefore a more rapid flame propagation and higher resulting pressure. It is obvious from this that if the amount of burned gas in the cylinder is changed the ignition point should also be changed in order to obtain the best results. It is even more important to have a perfectly combustible mixture; that is to say, to have just the right proportion of air and gasoline vapor in the cylinder. For certain compression pressures and certain fuels there is one proportion at which the flame propagation is highest. Even slight deviation from

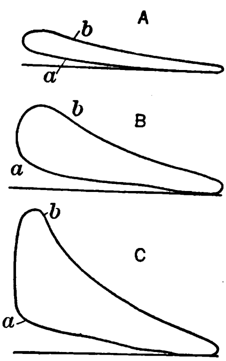


FIG. 10.

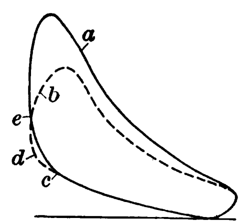


FIG. 11.

this proportion will make a marked difference in the velocity of propagation, the power and the thermal efficiency of the engine.

The higher the compression the smaller the quantity of gasoline vapor required to obtain the best results, and the later the point of ignition. If the point of ignition is not advanced when the charge consists of an excess of air or of gas, there will be an additional delay in the flame propagation, besides the reduction in the height of the curve as seen in Fig. 12. Here curve *b* shows an excess of air, and curve *c* an excess of gas. When the mixture is diluted the temperature will have to be increased before ignition of the total charge can take place, as the mixture is not perfect. When some portion of an imperfect charge is ignited, the heat from the combustion of that portion will first have to heat the surrounding layers before it can ignite them; this accounts for the increased time required for the combustion of the total charge. Curve *c*, Fig. 12 shows that some layers of the gas

are ignited very late, which explains the reason why the curve is not following the adiabatic law, and that at the end of the stroke the pressure, and therefore the temperature, is higher than that of curve *a*, which had a perfect mixture. It accounts also for the fact that when an excess of gas is used engines heat up and set the water in the radiator boiling. So far we have only considered the losses in the combustion line resulting from imperfect mixtures or weak compression, and we have seen that by shifting the point of ignition the losses can be decreased to a certain extent. Even more important, however, is to time the point of ignition correctly when the mixture is perfect, as in this case the resulting pressure will be higher.

We have seen from the foregoing that the flame propagation is not instantaneous; for this reason it is clear that variable piston speeds require different points of ignition to have the maximum

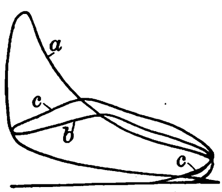


FIG. 12.

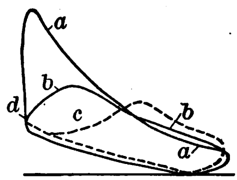


FIG. 13.

rise in pressure occur at the end of the stroke or slightly after dead center. If the point of ignition of an engine were adjusted to a speed of, say, about 400 r.p.m., and the mixture perfect, the combustion curve would rise to about four times the compression pressure (see curve *a*, Fig. 13). If the point of ignition be kept the same and part of the engine load be removed, so that the speed increased to 800 r.p.m., the indicator diagram would be like curve *b*; the dotted curve *c* shows a diagram that would be obtained at a speed of 1500 r.p.m.

The same type of curve is obtained if the engine speed is kept constant but the point of ignition retarded. It is evident by looking at curve *c* that the pressure, and therefore the temperature, is higher at the end of the stroke, which shows that late ignition will also heat up the engine. From curve *c* it can also be seen that the compression pressure is higher than that of curve *a* and *b*, the reason being that at higher speed the proportion of gas leaking past the valve seats and piston rings is smaller, and also that less heat is lost or given up to the cylinder wall during

compression. In this engine, if the point of ignition were advanced properly with the increased speed, the peak of the combustion curve would have risen above that of curve *a*, as seen by the dotted curve *e*, Fig. 14. This clearly shows that an increased speed will, under "favorable circumstances," give an increased compression and increased power per explosion. However, if the mean temperature of the cylinder wall were higher than the temperature at the end of compression, and the leaks very small, at low piston speed the compression would be higher than at higher speed, because a certain amount of heat of the cylinder wall would be abstracted by the gas during compression, thereby increasing the pressure, and this is sometimes the case in practice. Nevertheless, the expansion curve would have been better at higher speed, as we shall see later.

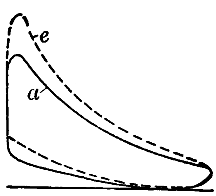


FIG. 14.

The most important reason, however, why the compression pressure in many cases is lower at higher speeds (the clearance being the same in both cases) is due to the increased loss of charge or suction resulting from the increased fluid friction through the valve and the inlet passage. Thus the importance of providing large valves and valve passages as a means for increased power from a given engine cannot be too strongly emphasized, for it means a larger volume of fresh charge, a higher compression, a quicker combustion curve, and therefore a higher speed and a better expansion curve (see chapter XVII on manifolds).

It was mentioned above that a higher compression was obtained at higher speed when the cylinder wall temperature was lower. This, however, does not imply an increased efficiency, but simply that there were greater wall losses at lower speed, and if the cylinder wall temperature were raised the fuel efficiency would have been increased.

WALL COOLING

The effect of excessive wall cooling upon the combustion curve is clearly demonstrated by Fig. 15. The five different curves of this figure show different temperatures of the cylinder wall. Such curves are obtained from the indicator if the water temperature in the jacket is suddenly lowered, and the greater the surface

of wall exposed to the hot gases (as, for instance, in large valve pockets) the lower the peaks. From these curves we can also draw the conclusion that by advancing the ignition more power could have been obtained from the charge; first, because the peak would have been higher, and, second, because the expansion curve would have dropped toward the end below the point indicated at *f*. Hence the proper point of ignition depends also to some extent upon the temperature of the cylinder wall, and therefore automatic spark timers which are governed by the speed of the engine are not absolutely correct.

When a cold engine is started the indicator diagrams would be similar to the curves of Fig. 15, showing that as the temperature of the wall increases the peaks will rise gradually from *a* to that of curve *e*. Another point worth noticing in this diagram is that the compression pressure is the same for all curves. Surely, it might be inferred that the compression should increase as the wall temperature increases. This would actually occur if the temperature of the inlet pipe and inlet passage were the same. However, from the diagram we can tell that this is not the case in the engine under consideration, as here the volume of fresh charge is somewhat larger when the wall is cold than when it is hot, and so the gain in charge is neutralized by the loss through the jacket. In some designs we find that by keeping the walls at a low temperature we can obtain higher peaks and more power from the engine; however, the thermal efficiency is nevertheless lower as the losses through the walls are greater.

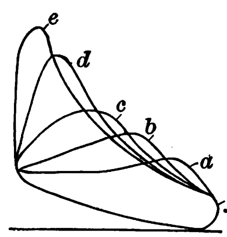


FIG. 15.

EFFECTS OF THROTTLING

Throttling an engine—that is to say, reducing the volume of the charge—will reduce the compression pressure, thereby retarding the combustion and reducing the height of the peak. Curves *a* and *b*, Fig. 16, illustrate the effect of throttling. As the compression line of curve *b* crosses the atmosphere at the middle, *x*, of the diagram, the volume of charge is one-half, and, naturally, the compression pressure and the height of the peak are lower. Throttling, however, also lowers the efficiency of the engine, as,

in the first place, there is a loss represented by the lowered suction line, and, secondly, a decreased compression, as mentioned before, reduces the thermal efficiency of the engine.

Another point worth noticing on Fig. 16, is that curve *b* shows the effect of retarding ignition, in this case chiefly due to the lowered compression; by advancing the ignition the area of the curve could have been somewhat increased, as seen from dotted line *c*. This shows the importance of changing the spark lever with a corresponding change in the throttle lever in automobiles. While it is aimed to have the compression as high as possible, there is a point above which an increase in compression will not give more power, owing to the increased losses on compression and especially the heat units lost through the cylinder wall. With gasoline vapor this point is seldom reached, so the designer need not consider it. The greatest drawback to high compression

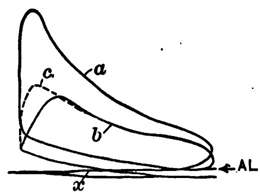


FIG. 16.

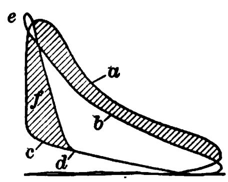


FIG. 17.

in automobile engines is the increased temperature, causing self-ignition of the charge, which has the same effect upon the combustion line as early ignition.

EFFECTS OF EARLY IGNITION

If the ignition occurs too early, the combustion line will naturally appear before dead center, thereby decreasing the power correspondingly. Figure 17 gives two curves *a* and *b*, the former with a correct point of ignition *c*, and the latter with one too much advanced, *d*. From *d* the pressure rapidly rises to *e*, the end of the stroke.

When the piston passes dead center the pressure line drops below the ascending line and below the expansion line of curve *a*. The top of curve *b* will vary greatly with the piston speed and the dilution of the charge. In this case the piston speed was not too high, and the mixture had a rather low velocity of flame propagation, which accounts for the loop at the top of the curve.

It indicates that the temperature, and therefore the pressure, decreased proportionally more rapidly than the piston passed dead center. Excessive cooling might also cause this loop to occur at slightly advanced ignition points.

The loss in power arising from too early ignition is represented by the shaded area of curve *a*, which shows the excess of power over that of curve *b*. The shaded area *f* indicates the power directly opposing the motion of the piston. If this area, added to that of the compression curve (see Fig. 18) equals about 85 per cent. of the area left on the diagram, it will usually stop the engine at no load, the remaining 15 per cent. of the area being neutralized by the power lost on suction and compression, and by increased friction losses. The weight of the flywheel, however, has the controlling influence in this matter, for a heavy flywheel will carry the crank around a number of times when this shaded area reaches even larger proportions.

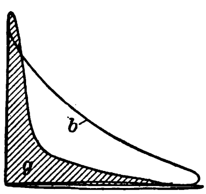


FIG. 18.

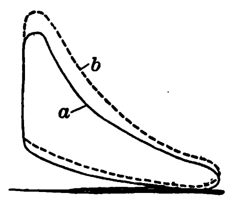


FIG. 19.

EXPANSION CURVES

The height of the expansion curve from the atmospheric line is the curve which furnishes the power in an engine and it will depend on

1. The initial volume of charge.
2. The condition of the mixture.
3. The compression pressure.
4. The point of ignition.
5. The speed of expansion.
6. The losses through the cylinder wall.

Items 2, 3, and 4 have been covered before. As regards the volume of charge, its importance is plainly manifest from Fig. 19. Both curves *a* and *b* were obtained with the same piston speed, the best mixture and the best point of ignition. Curve *a* was taken in the ordinary way (compression commencing from 13 lb. absolute), and its shape corresponds to good average

practice. When curve *b* was taken the charge had been slightly increased and compression began nearer atmospheric pressure (about 13.5 lb. absolute). Although the difference at the end of the suction lines or the beginning of the compression lines was only about ½ lb. the increased power as manifested by the increased area was astonishing. It might be noted that the compression of curve *b* was only a little higher than that of curve *a*, and that the main reason for the better expansion curve was the increased volume of fresh charge.

By increasing the valves and valve passages, we naturally obtain a larger volume of charge, but, on the other hand, an increase in valve size means larger valve pockets, and the latter are conducive to greater heat losses. (In some of the latest designs, four valves are used in the cylinder head—two inlet and two exhaust valves—and in this manner large pockets are eliminated.) Theoretically, the greater the area of cylinder wall exposed to the hot gases, the greater are these losses, as the expansion curve will drop more rapidly. By increasing the cylinder wall temperature these losses can be reduced to a great extent; but if the temperature is too high we are confronted with mechanical difficulties, as the lubrication of the piston will fail, when the increased friction will more than counteract all gains or cause the piston to bind. From practical tests made it was ascertained that the best results are obtained when the jacket water is near the boiling point. (In different designs, however, this may vary somewhat; likewise will this temperature depend upon the quality of the lubricating oil and the piston speed.)

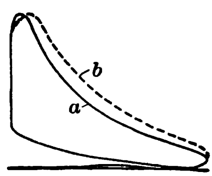


Fig. 20.

In order to obtain the highest efficiency, the difference in the temperature of the water entering and leaving the cylinder should be a minimum. The time constant during which the hot gases come in contact with the wall also affects the thermal efficiency, as evidenced by Fig. 20. Here both curves were obtained under the most favorable conditions in regard to ignition and the charge, except that curve *a* was taken at a speed of 300 r.p.m. and *b* at 1000 r.p.m. The compression was the same in both cases, but the ignition, as can be noticed, had to be advanced for curve *b*, the same as if the flame propagation were slower. This, however, is not the case; the curve is more rounded simply

on account of the higher piston speed, and it expands at higher pressures because of smaller thermal losses.

In practice the expansion curves are sometimes very irregural in shape, which may be due to imperfect mixtures of the gases, some layers igniting during the expansion stroke; however, the inertia of the moving parts of the indicator is frequently responsible.

EXHAUST LOSSES

By looking at the different diagrams it is noticed that near the end of the stroke the expansion curve drops very rapidly below the adiabatic line, which naturally indicates a certain loss in area and, therefore, in power. This sudden drop is due to the opening of exhaust valve before dead center, in order to allow a certain amount of burned gas to escape before the piston returns, otherwise there would ensue a great loss in power due to the high back pressure. If the exhaust valve area were as large, or nearly as large, as that of the cylinder, there would be no appreciable back pressure, and in this case the exhaust valve would not have to be opened before the end of the stroke. In practice all valve

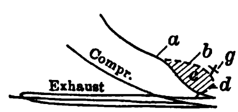


FIG. 21.

passages are constricted and the exhaust valve should be so timed that the loss resulting from an early opening, plus the loss arising from back pressure, should be a minimum. Here, too, the indicator diagram is the surest guide for setting the valve. If the exhaust valve is opened too soon the end of the expansion curve will drop to nearly atmospheric pressure before the end of the stroke is reached (see Fig. 21). Here *a* indicates the point of opening, and the shaded area *c*, minus the area *d*, shows the gain in power if the release had taken place at *b*. It is true that even with such an early opening there is a certain loss, due to back pressure on the exhaust stroke, but this always occurs, for the reason before mentioned. In this case the valve area and valve lift were such that the exhaust gas escaped at a velocity of about 5000 feet per minute. Had the valve been smaller it would have been necessary to lift the valve earlier than at *b* in order to keep the loss down.

If the point of opening *a*, occurs too late the loss arising from back pressure is represented by the shaded area of Fig. 22, and it is due to a late exhaust opening. Although the small

shaded area at f shows a loss in power which could have been eliminated by opening the exhaust somewhat later, this loss is very small compared with that prevailing during the exhaust stroke, as shown by the shaded area above the atmospheric line.

Thus the exhaust valve area has an important bearing upon the valve timing and a late release is always conducive to high back pressure losses. Take, for instance, the diagram, Fig. 21, had the point b been retarded to g the back pressure at the beginning of the exhaust stroke would have risen as high as that of Fig. 22, but if the valve area had been large it would have dropped toward the end of the stroke, as shown by the dotted line c.

An ascending exhaust line Fig. 23, on the other hand, indicates an abnormally high gas velocity through the valve, and when such a curve is obtained the valve area should be enlarged. Of course, the valve lift and valve passages are just as important as the area. The shape of the cams also influences the timing of the valve, for if the lift takes place too slowly proper allowance has to be made

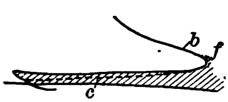


Fig. 22.

Fig. 23.

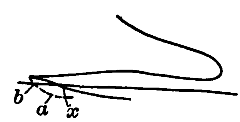


Fig. 24.

in the setting. A high back pressure may also result from the muffler, if it has either too small a capacity or too small passages, and also if the exhaust pipe area is too constricted, or if there are sharp bends in the pipe, thereby increasing the resistance.

The closure of the exhaust valve is another important matter; if occurring too early an excess of burned gas will remain in the cylinder (there would also be a rise in pressure), while if kept open too long some of the burned gas will re-enter the cylinder during the suction stroke. By looking at the last two diagrams, it is seen that at the end of the exhaust stroke the pressure in the cylinder lies above atmospheric pressure; therefore there will be no danger of exhaust gas being sucked back until the piston on its return stroke passes the point x, Fig. 24, where the suction line crosses the atmospheric pressure. In this diagram the exhaust valve was closed at the end of the stroke, and in order to prevent exhaust gas from passing through the inlet valve the latter had to be kept closed until the pressure dropped to that of the atmosphere. The pressure at the end of the exhaust stroke

indicates the excessive amount of burned gas, and we have learned before of the harmful effect this has upon the fresh charge. Any means reducing the volume of the burned gas will result in an increased engine power.

SCAVENGING

At the opening of the exhaust valve, the pressure in the cylinder being considerable, the burned gas will escape at a high speed, and during the exhaust stroke, the piston, moving also at a high speed, will assist in forcing the burned gas through the comparatively long exhaust pipe. A volume of gas moving at such a velocity will naturally have some inertia, and if the exhaust valve is kept open it will be found that this inertia will draw after it a portion of the burned gas left in the cylinder at the end of the stroke, thereby reducing the pressure in the cylinder. This action, called "scavenging," is often employed in automobile engines. The inertia of the exhaust gas when the release of the exhaust valve is very early may be seen in Fig. 25. Release in this case occurs at *c*; at *y* the exhaust pressure line crossed the atmospheric line, and the subsequent drop to *f* below atmosphere was directly due to the vacuum created by inertia of the escaping gas.

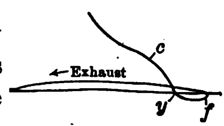


FIG. 25.

If the exhaust valve in Fig. 24 were kept open up to point *a* on the suction stroke, the dotted curve would have been obtained, showing a large decrease in the volume of burned gas. Sometimes the inertia of the exhaust gas is also employed to assist in setting the fresh charge in the inlet pipe into motion toward the cylinder at the beginning of the suction stroke. In this case the inlet valve may open at *b*, so that both valves are open from *b* to *a*. In high-speed engines the velocity of the exhaust gas is higher, other things being equal, therefore the inertia is higher; but so is also the back pressure. If with a given size valve the engine speed were steadily increased a point is reached when an increase in speed is accompanied by a decrease of power, owing to the increased back pressure and the decreased volume of charge.

When setting the exhaust valve, the same as when setting the inlet valve, the engine should be run, not at a maximum speed, but at the speed at which the engine will mostly run, as the highest efficiency at normal speed will be the most economical.

In multi-cylinder engines indicator diagrams should be taken from every cylinder (see Chapter on Engine Testing), for frequently an improper valve setting of one will affect the diagram of the other. For instance, too early an opening of the exhaust valve of one cylinder may raise the exhaust curve at the end of the stroke of another cylinder. A similar effect may be obtained by too late a closure of one exhaust valve, in which case the exhaust gas from another cylinder may blow back and indicate an increased pressure at the beginning of the suction line.

CHAPTER V

CYLINDER DIMENSIONS

BORE AND STROKE, THICKNESS OF CYLINDER WALL, WATER JACKET, ETC.

For a number of years automobile builders used engines in which the bore was about equal to length of stroke. It was reasoned that with such a short stroke there would be a larger number of explosions per minute for a given piston speed, and that this means more power. Afterwards, some manufacturers increased the length of stroke and found that the longer stroke not only gave more power, but rendered possible a higher piston speed per minute.

No doubt, the chief reason for the increased power is that with a longer stroke, there is more time (at a given piston speed) for the fresh charge to enter the cylinder, and as we have seen from previous chapters, a small additional quantity of charge will make a considerable difference in area of the expansion curve of the engine.

It takes some time to overcome the inertia of the fresh charge, or to set it into motion to flow into the cylinder, and in the longer stroke it has, of course, more time to flow into the cylinder, and the inertia of the charge does not have to be overcome so often for a given piston speed. This also holds good of the inertia of the reciprocating parts, *i.e.*, the piston and the connecting rod; while the connecting rod in the long stroke motor is somewhat heavier, the direction of motion of the reciprocating parts does not have to be reversed so often, which means less losses due to inertia. As regards the fresh charge, for a given cylinder volume its proportion is larger with a longer stroke for the reason stated, and this proportion of charge to volume of piston displacement is termed the *volumetric efficiency*.

It is evident (see Chapter IV) that if the proportion of fresh charge to burned gas is larger, the explosive mixture will be purer, thus resulting in a higher expansion pressure and in a quicker flame propagation. In this manner, the volumetric

efficiency affects the purity of the charge as well as its quantity and thus influences the speed of the engine as well as its power, while the reduced inertia of the reciprocating parts means reduced losses.

It is customary to make the proportion of stroke to bore larger where the diameter of the cylinder is smaller than where it is larger. The average practice, at the present time, is to make the proportion of stroke to bore approximately as follows:

For a bore of from 3 inches to $3\frac{7}{16}$ inches, the proportion is 1.55 to 1.

For a bore of from $3\frac{1}{2}$ inches to $3\frac{15}{16}$ inches, the proportion is 1.45 to 1.

For a bore of from 4 inches to $4\frac{7}{16}$ inches, the proportion is 1.35 to 1.

For a bore of from $4\frac{1}{2}$ inches to $4\frac{15}{16}$ inches, the proportion is 1.25 to 1.

For a bore of from 5 inches up, the proportion is 1.15 to 1.

One reason for having the proportion of stroke to bore smaller for larger engines, is no doubt that increasing the stroke means a direct increase in the size, and thus also the weight, of the cylinders, the connecting rods, cranks of the crankshaft, and the crankcase, and indirectly in the weight of the fly-wheel, on account of the smaller number of explosions per minute for a given power. In a large engine the height and weight would be excessive if the stroke were made 1.5 times the bore.

Another reason is probably that the purchaser of a large car is not so much interested in "miles per gallon" as the owner of a smaller car, but he would object to an abnormally high hood necessitated by excessive engine height.

The first consideration in the design of a motor is the power it is to furnish. (See chapters on Horse Power, Tractive Factor and Torque.) Practice in the past has shown that a torque of about $6\frac{3}{4}$ pound-inches (lb.-in.) is obtained per cubic inch of piston displacement. The formula for finding piston displacement in cubic inches is $\pi r^2 L$, where r is the radius of the cylinder, in inches, and L the length of the stroke, in inches; and $\pi r^2 L = .7854 D^2 L$, where D is the diameter of the cylinder, in inches.

Take for instance the "B" type Quartermaster engine manufactured for the United States Government, where a torque of 2800 lb.-in. (pound-inches) was decided upon as necessary. In

order to get the requisite volume of piston displacement we divide 2800 by 6¾ which = 415 cubic inches. By trying the proportion of stroke and bore according to the figures previously mentioned we find that a 6-inch stroke and a bore of 4¾ inches has a volume of piston displacement of 425 cubic inches and this is in the ratio of stroke to bore of approximately 1.25 to 1. It might be stated that 4¾ × 6 inches was the bore and stroke decided upon in the type "B" Quartermaster truck engine, of which many thousand were built and which may be said to represent the latest American practice in truck design.

CYLINDER WALL THICKNESS

When determining the dimension of the cylinder, the bore and stroke having been established, we next consider the maximum pressure per square inch resulting from the explosion, and this is usually taken at 400 lb., although in practice this pressure is ordinarily not as high. The tensile strength of cast iron varies from about 10,000 lb. to 30,000 lb. per square inch; ordinary good close-grained cast iron of which cylinders are usually made, has a tensile strength of about 22,000 lb. per square inch.

To determine the thickness of the cylinder wall metal, we may proceed as follows: First, note the force which tends to burst the cylinder per unit length and this = PD, where P is the maximum pressure per square inch, and D the inside diameter of the cylinder.

Opposing this force, we have the strength of the cylinder wall, which is the thickness t of the cylinder wall multiplied by its tensile strength T. The metal at any two diametrically opposite points equally resists this force, therefore, $2tT$ = the cylinder wall strength. This is illustrated in Figure 26. Supposing a pressure of 400 lb. per square inch is acting in the direction of the arrow, and distance A and B are one inch each; the strength of the metal on both sides, c and d, will tend to resist rupture. If this metal has just sufficient strength to withstand the explosion pressure, we have $\frac{2tT}{PD} = 1$. In addition to the thickness of metal actually required, it is necessary to have several times the strength needed, and this is called the factor of safety.

This factor of safety is required for several reasons—jars

and vibrations; the fact that the tension in the metal is not uniform but is greater at the inside of the cylinder wall than at the outside; and the strength of the metal deteriorates in use.

We take a factor of safety of 5, *i.e.*, make the cylinder wall 5 times as strong as actually necessary; our formula thus becomes $5PD = 2tT$, therefore, the metal thickness $t = \frac{5PD}{2T}$, and by substituting the values before given, we have $t = \frac{5 \times 400 \times D}{2 \times 22{,}000}$ $= .045D$. An additional $\frac{1}{16}$ inch should be allowed as the metal is not absolutely uniform in thickness and the casting is liable to have small blow holes, therefore, the formula becomes $t = .045D + .063$.

It is evident, the larger the diameter, the greater the stress in the metal, although I have had students inquire why this

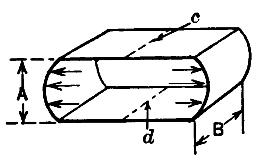


Fig. 26.

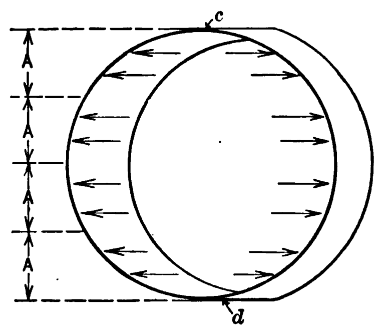


Fig. 27.

should be so, since the pressure per square inch remains the same. This can easily be understood when looking at Fig. 27. The area inside the cylinder is greater (there are many square inches with a pressure of 400 lb. on each) and therefore the total tension at c and d is greater; this tension increases with the diameter and the pressure.

When, in the casting of cylinders the cores are specially well set and a good, hard, close-grained metal is used, the thickness of cylinder wall is frequently made somewhat less; ordinarily, however, the foregoing formula will be found to give satisfaction, and to correspond with good American practice.

In high-speed engines it is not advisable to make the cylinder wall too thick, as the stress set up in the metal due to the difference in temperature on the inside and the outside of the wall is

liable to crack the cylinder. Whenever possible cylinder walls should also be made of even thickness all around. This means an even shrinkage and an even expansion when the cylinder becomes hot, and avoids breakage from shrinkage which sometimes occurred in the earlier days of the automobile industry. The only portion of cylinder wall which is made thicker is where it is jointed to the flange as practice has shown this to be a weak spot. This is shown on Fig. 29.

It was shown before that (see Wall Cooling, page 24), theoretically, the smaller the surface of the combustion chamber and cylinder wall in contact with the hot gases, during explosion as well as during expansion, the higher will be the efficiency of the engine because less heat is lost through the walls. The greatest volume contained within the smallest surface of wall is in a spherical form. The surface of a sphere is $4\pi r^2 = \pi d^2$; its volume $= \frac{4}{3}\pi r^3 = \frac{\pi d^3}{6} = .5236d^3$. The surface of a cylinder (closed on all sides) $= 2\pi rh + 2\pi r^2$, while the volume of a cylinder $= \pi r^2 h = .7854d^2h$. (h = length of cylinder, r the radius, and d the diameter.)

It is impractical to make the cylinder a sphere or even to make the piston head concave as shown in Fig. 28, which approaches the shape of a sphere. This shape of piston head has the tendency of becoming over-heated, for the piston head proper has a greater surface exposed to the heat of combustion than is the case with the ordinary flat piston head, and as the cooling of the piston is chiefly effected through the cylinder wall, this type would get over-heated. For this reason, the best practical form of combustion chamber is to make the cylinder head concave and the piston head flat, but even this is not often possible to do in practice, except with overhead valves. Ordinarily when the valves are located in a pocket or on opposite sides of the cylinder, the compression space would be too large (hence the ratio of compression too low) if the head were made spherical as shown; in such cases it frequently has to be brought down very low to obtain the required compression.

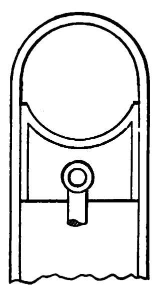


FIG. 28.—Spherical combustion chamber.

The explosion chamber is subjected to very high temperature changes in the present-day high speed engine and this necessitates

the provision for a uniform expansion of the walls. Unless this is done distortion of the valve seats may occur.

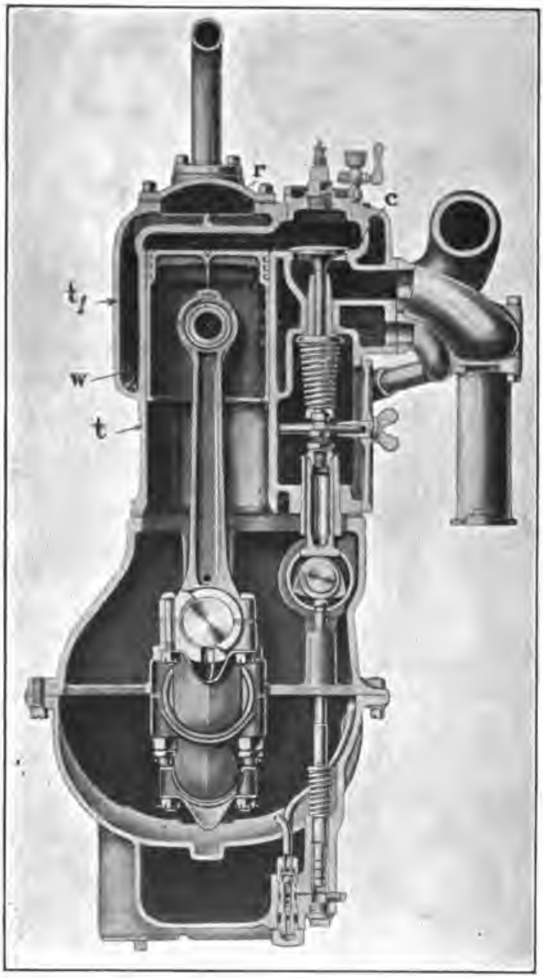


FIG. 29.—Model E, continental engine.

The cylinder barrels should be isolated from each other by a wall of water. In some of the earlier engines where the cylinders were cast together without a water space between the cylinders, the uneven expansion caused the cylinders in time to become distorted.

Sometimes a rib r (Fig. 29) is placed on the outside of the ex-

plosion chamber extending radially to the jacket. These ribs are made about one-fourth of an inch thick. When designing the valve pockets it is important that sufficient clearance *c* be provided around the valve seat to freely permit the passage of the gases. The clearance all around the valve should in no case be less than one-fourth of an inch. In order to reduce the friction of the gas as much as possible, sharp angles should be avoided in the combustion chamber. All passages should be well rounded and smooth, to have the resistance a minimum.

In the foregoing chapters it was pointed out that the smaller the surface area of the combustion chamber in contact with the hot gases, the smaller will be the theoretical loss in the engine. Regarding the length of the inlet passage, if all the surfaces are smooth and there are no sharp turns, the losses are very small. (Sometimes a longer inlet passage will give a more perfect mixture.) It is of more importance however, that the exhaust passage be as short as possible to expel the burned gases without much resistance. Discharging all the exhaust passages from the cylinders into the same continuous header may also cause uneven expansion of the cylinder walls and distortion.

The water outlet should be at the highest point of the cylinder to prevent air pockets. The water inlet is placed at the lowest point although this practice is not always followed. A tap should be provided to permit the removal of all the water from the cylinder by gravity.

Detachable cylinder heads, *y* (Fig. 30) as used with the block type design, facilitates machining of the cylinders, but extra care is necessary to make the joint between cylinder head and cylinder pressure tight. Steel studs made of chrome-nickel steel are recommended to bolt the head to the cylinder casting, which latter should also be provided by reinforced bosses *b*, tapped for the studs. The length of the tapped hole should be at least three-fourths of an inch, except for very small cylinders, and the diameter of the steel studs may be made 3/8 inch in diameter for a 2½-inch bore; 7/16 inch for a 3-inch bore; ½ inch for a 4-inch bore; and 9/16 inch up to a bore of 5 inches. The gasket used in the joint should be copper asbestos, preferably with the border copper-lined, to prevent the asbestos fabric from being squeezed out. In some air cooled engines no gasket is used, but one of the surfaces of the joint is covered with very small grooves presenting the appearance of a rough surface. The leaks, pass-

ing through the joint, deposit carbon there, making the joint pressure tight, ordinarily, in two or three hours. The base of the cylinder is frequently reinforced with long, thin ribs, placed between the bosses at the cylinder base and extend to the water jacket. By making the top of the cylinders absolutely level, *i.e.*, the flanges of water pipe, of valve cap, spark-plug base and priming cup boss, the machining operation is greatly simplified and is accomplished at one operation. At

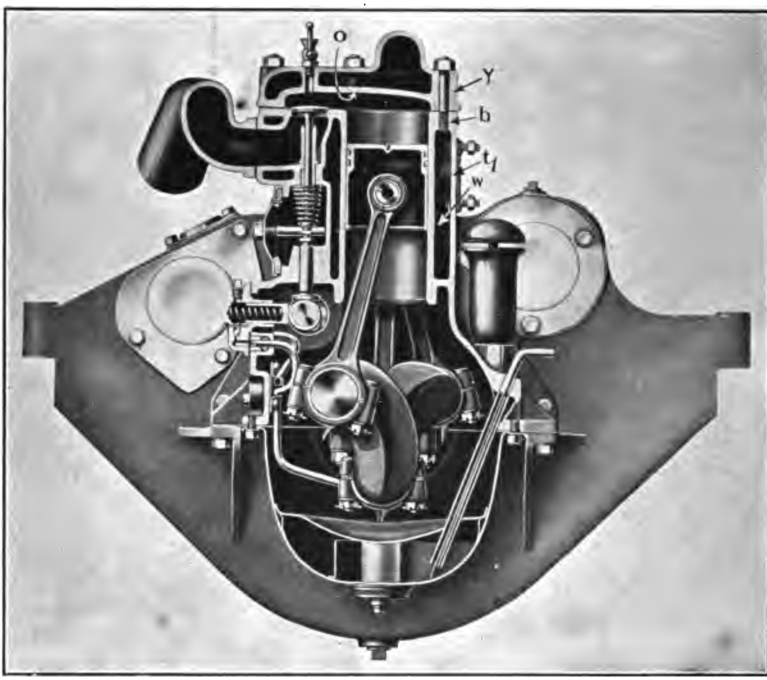


Fig. 30.—Model 7W, continental engine (with detachable head).

the sides of the cylinder it is likewise advisable to have the inlet water flange and the inlet and exhaust pipe flanges in the same plane, if they are all on the same side.

Figure 29 shows a sectional view of the Model "E" Continental engine, which has been a successful model for several years. Figure 30, Model 7 "W," is a later design with detachable cylinder head.

Figure 31 shows the individual, detachable cylinder heads of the Hercules Motor, which is especially built for heavy duty.

Note that the water circulates between the cylinders and the heads through the oblong openings *V*, while through the round holes pass the bolts which fasten the heads to the cylinders.

Fig. 31.—Hercules motor with detachable cylinder head.

WATER JACKET

The water jacket extends, as a rule, all around the cylinder head, and slightly below the lowest point of the position of the piston (see Fig. 29). In the design shown in Fig. 30 it is brought down almost to the bottom of the cylinder, to insure an even expansion and contraction of the entire cylinder wall. The depth of the water jacket is, as a rule, somewhat narrower at the bottom than at the top, and there is a substantial increase in depth around the cylinder head and exhaust valve. The depth w of the water jacket is usually made $\frac{7}{16}$ inch for an engine having a bore of 3 inches, and is increased by $\frac{1}{16}$ inch in depth for every $\frac{1}{2}$ inch increase in bore. This may be arranged in a formula as follows and holds good for cylinder diameters from 3 inches to 6 inches: $t = .437 + (b - 3).125$, since $\frac{7}{16} = .437$, and $(b - 3)$ is the increase in diameter above 3 inches, b being the cylinder bore.

The thickness t of the jacket wall may be made $\frac{1}{8}$ inch for a cylinder with a bore of $2\frac{1}{2}$ inches, adding $\frac{1}{32}$ inch for every one inch increase in bore, thus for a bore of $3\frac{1}{2}$ inches, the jacket would be $\frac{5}{32}$ inch thick, etc.

In these dimensions, there may be a slight variation, depending

upon the quality of the cast iron, but these figures are a good average of the latest designs.

The water jacket is usually made somewhat smaller where the valves are located, to accommodate the spring and the valve seat; however, near the valve stem it is usually increased as shown in Figs. 29 and 30. In multi-cylinder engines, the depth of water jacket is usually smaller between cylinders (where cylinders come closest together) than on the outside, to avoid unnecessary length of engine.

After the cylinders are rough-bored, a number of manufacturers set them aside for ageing, to eliminate distortion, due to machining strains. A rigid test under water pressure is given the cylinders before and after they are machined. Afterward they are finish-bored and carefully ground to standard size.

It is very important that the inside of the water jacket be thoroughly cleaned and to remove all fins, coarse sand, or other foreign matter.

Fig. 32.—Hercules cylinder block. Section through intake passage.

The gas intake passage in Fig. 30 is contained entirely within the cylinder casting (the same as in the construction shown in Fig. 32) which gives the effect of a water-jacket manifold, and it also eliminates pipe connections on the outside.

In the early days of the Motor Vehicle industry the cylinders were cast singly and bolted on to the crankcase.

One of the more recent tendencies in engine design is to cast the cylinders en bloc, and to have separate heads attached thereto. With the separate cylinder heads, the user finds it more efficient and convenient to take off the head, scrape the carbon from the

cylinders, grind the valves; also he finds the removal of pistons and connecting rods facilitated.

The separate head offers important manufacturing advantages by enabling the upper half of the crankcase, or the crankcase entire, to be cast with the cylinders en bloc. This saves several machining operations, and by making it possible to properly support the core, it insures a uniform thickness in the cylinder wall, thus permitting such wall to be cast thinner without danger of defects. It is also a more economical construction, as the crankcase can be made of cast iron instead of aluminum, which is more expensive; by casting the crankcase en bloc with the cylinder, the heavy flanges or joints between cylinder and crankcase are obviated, thus saving about the same weight as is due to the use of cast iron instead of aluminum. With the detachable head, the entire combustion chamber *O* (Fig. 30) can be machined, thus insuring absolute uniformity in the volume of all the cylinders.

Mono-bloc castings are employed to-day for even the largest motors, but especially for the four-cylinder engines. However, many four-cylinder truck motors are cast in pairs. A large number of the sizes are cast in two sets of three; this relates especially to the greater horsepower six-cylinder engines which, on account of the size, is more difficult to cast in a unit with the crankcase. Other features made possible by the bloc casting are: only one inlet and one outlet water pipe for the entire engine; one inlet and one exhaust manifold cast integral with the engine (with the exhaust this is not done so frequently to prevent overheating of portions of the casting and thus distortion); the enclosure of every moving part, thus tending toward silent running, and making the appearance simpler; keeping the outside of the engine free from dirt and grease.

Figures 33, 33*a* and 33*b* show working drawings of the Buda 4¼ inches × 5½ inches cylinder casting. The inlet valves "*I*" of two adjoining cylinders, unite in a common passage but the exhaust valves *E* discharge separately. Figure 32 shows the "Hercules" Cylinder Block in section, being cut through the center of the intake passage which latter passes between the cylinders.

Figure 33*a* shows this is also the case in the Buda motor; the inlet passage *P* starts from the opposite side of the valves as shown in dotted lines. One object of this is to more thoroughly

vaporize the fuel before reaching the cylinder. The exhaust valve seats *S* are entirely surrounded by the water jacket to prevent over-heating, "*w*" shows the water jacket. Note the amount of water surrounding the valves and the valve guides which fit into openings "*G.*" Above the valves are the threaded apertures *H* for the valve caps, which latter are necessary for access to the valves; they are placed into their seats from above.

As a rule the spark plug is inserted in the center of one of the valve caps. In order to prevent overheating of the spark plug points, it is recommended that they be placed over the inlet valve, where the fresh incoming charge will cool them.

Fig. 33 plainly shows the space for the cap over each valve. The more thoroughly the valve caps are cooled by the surrounding water (see Figs. 33*a*, 35, 36 and 37), the less liability will there be for preignition caused by overheated spark plugs. In Fig. 37, two spark plugs are shown, one over each valve, as a number of tests carried out have disclosed the fact that ordinarily more power is obtained from an engine if more than one spark plug is provided, undoubtedly because by firing the charge at more than one spot, a more rapid flame propagation is obtained.

Of course, if there should be preignition due to an overheated plug (as a rule the plug over the exhaust valve, which is in the direct path of the hot gases during the exhaust stroke, is not cooled by the fresh ingoing charge during the suction stroke), instead of more power, the engine might give less power and run irregularly.

Sometimes a trouble of this kind is difficult to locate. Some engineers prefer to have the spark plug points just flush with the inside of the cylinder wall, so as to protect them somewhat from the heat of the combustion, others recommend they project into the combustion chamber in the path of the incoming fresh charge.

Mr. Sparrow (see paper read before the Society of Automotive Engineers, January 7, 1920) found that "to attain cooling of a spark-plug element, it is more effective to give free access to the cooling action of the incoming charge, than to sacrifice this effect to protect the element from the hot explosive gases."

The threaded opening at *K* is here used for a plug, which is tapped for a stud for the purpose of fastening down the water-jacket cover or the water manifold on top of the cylinders. See how *H* and *K* are surrounded by the water jacket. At

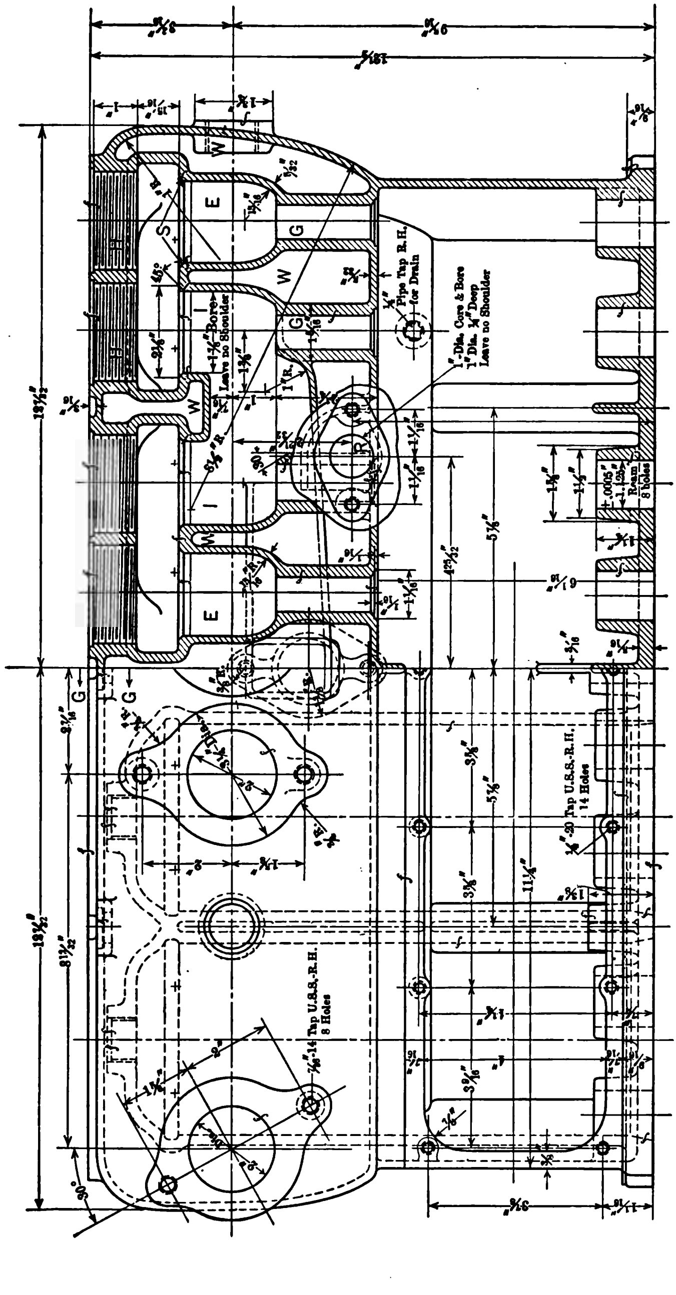


FIG. 33.—Buda $4\frac{1}{4} \times 5\frac{1}{2}$ cylinder casting.

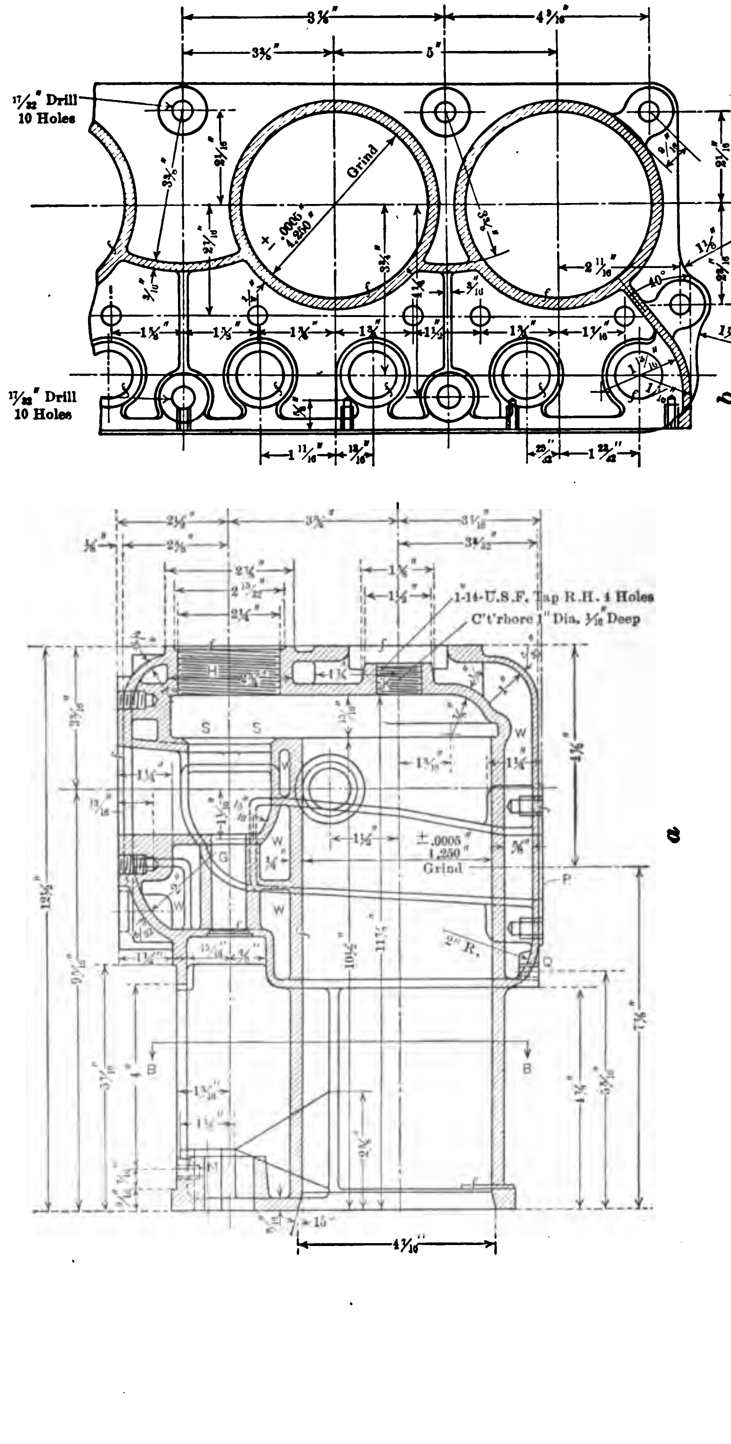


FIG. 33*a* AND *b*.—*a* shows section through center of cylinder and through exhaust valve; *b* section through B–B.

"*L*" the valve passage is tapered, decreasing in size toward the valve.

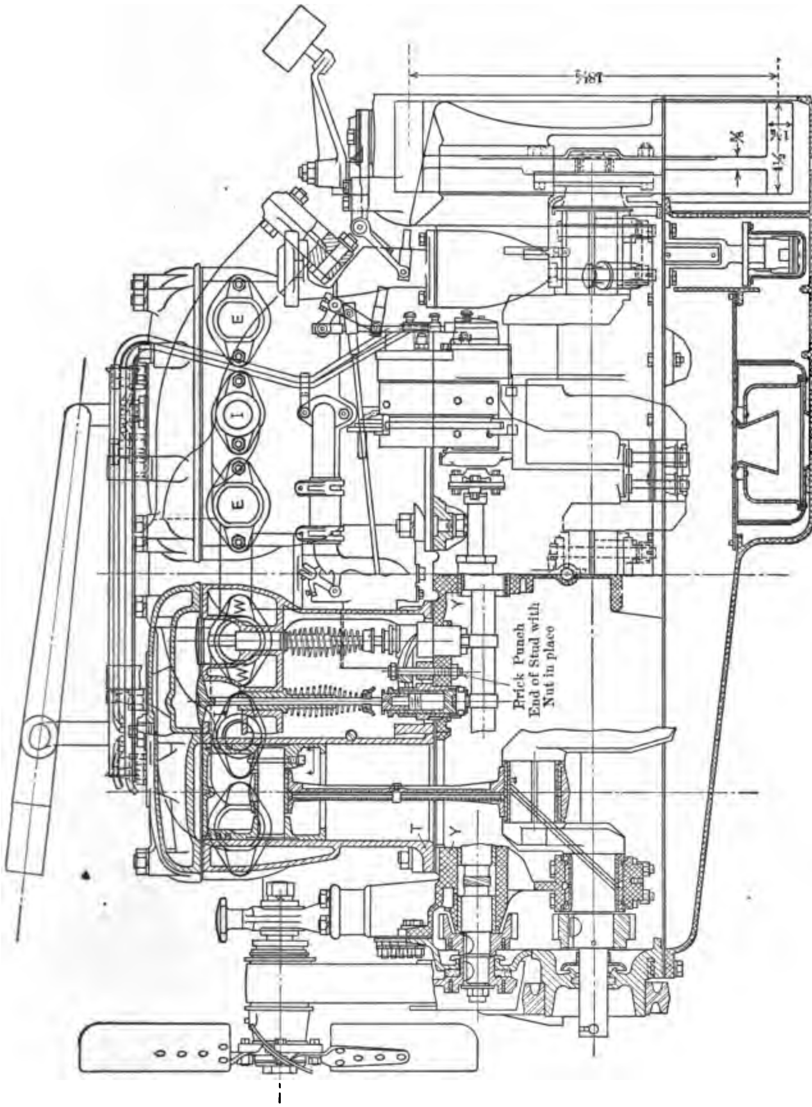


FIG. 34.—Class B, U. S. A. military truck engine $4\frac{3}{4} \times 6''$.

It has been found that in the case of an inlet passage if it is made narrower just before reaching the valve, the speed of the gas is increased (venturi tube effect); in the case of the exhaust

valve if the passage is broadened toward the outside flange, after leaving the restricted area, *i.e.*, if the gas is rapidly expanded, it has the effect of increasing the speed of the exhaust gas through

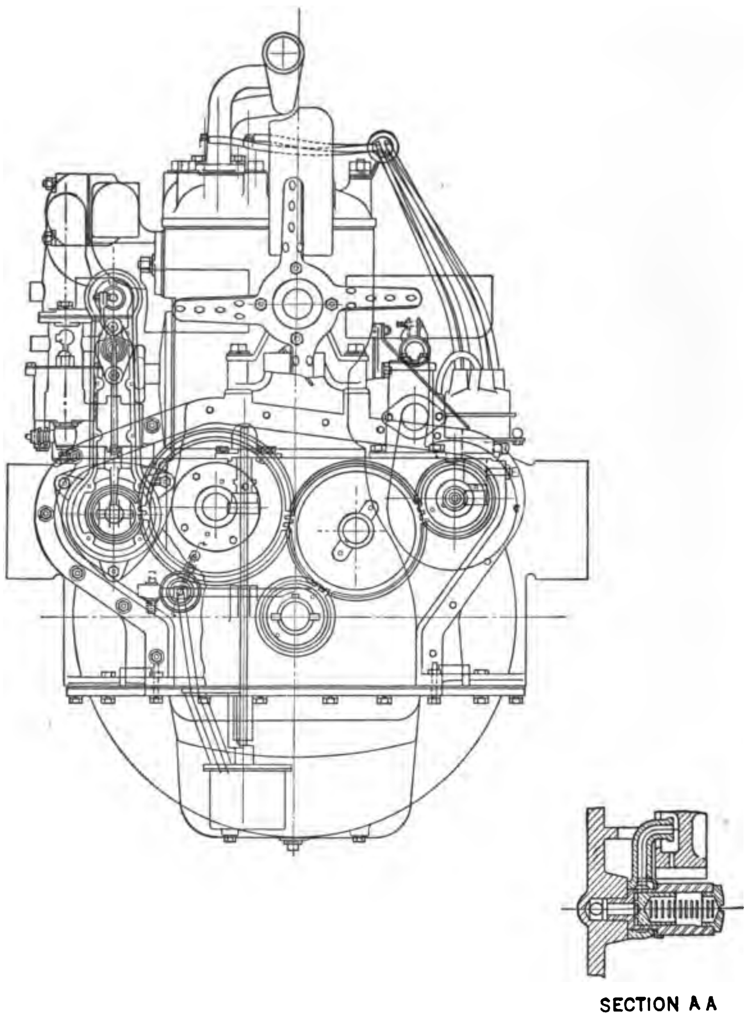


FIG. 34*a*.—Front view of Class B engine.

the valves. This is very important, for with the exhaust it means that the hot gases are eliminated at a more rapid speed, thus, giving off less heat to the cylinder wall. In the case of the

intake passage, an increased speed of the charge means a greater inertia of the in-rushing gas, and this has the effect of more completely filling the cylinders, *i.e.*, the volumetric efficiency is higher.

At "*M*" the push rod guides are inserted, but these will be discussed in a subsequent chapter. At "*Q*" provision is made for a plug to drain all the water from the cylinder; at "*R*" is

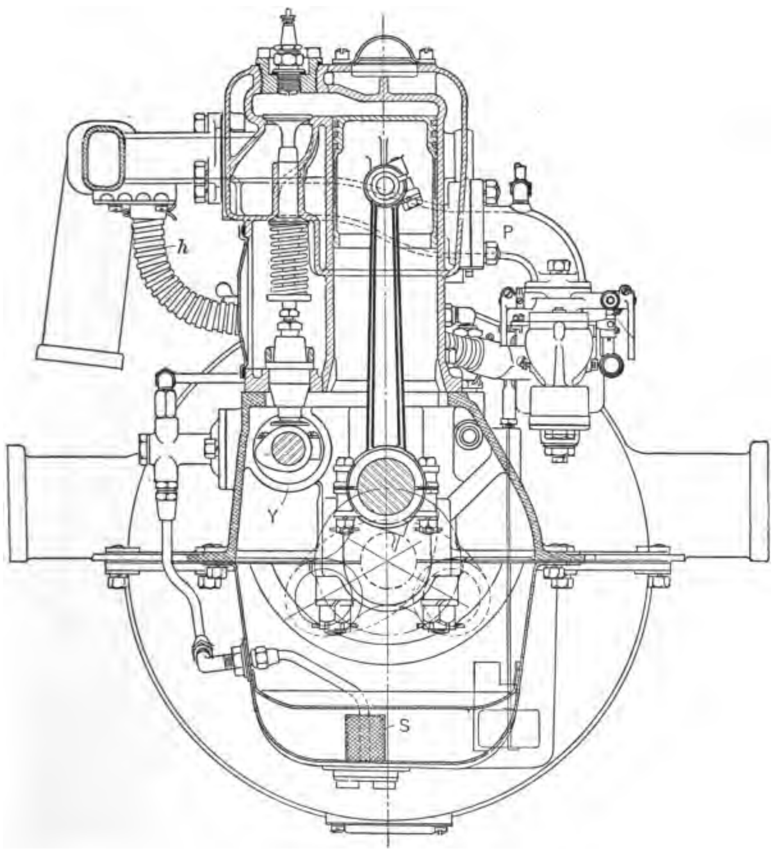


FIG. 35.—Haynes light-six motor. 3½″ × 5″.

the flange for water pipe connection coming from the pump. The water outlet is on the top of the cylinder.

Figure 34 shows a sectional view through the valves, and a view through the cylinder of the Type *B*, Quartermaster Engine of the U. S. Government. This motor has a detachable head and will be more fully described in subsequent chapters, but attention

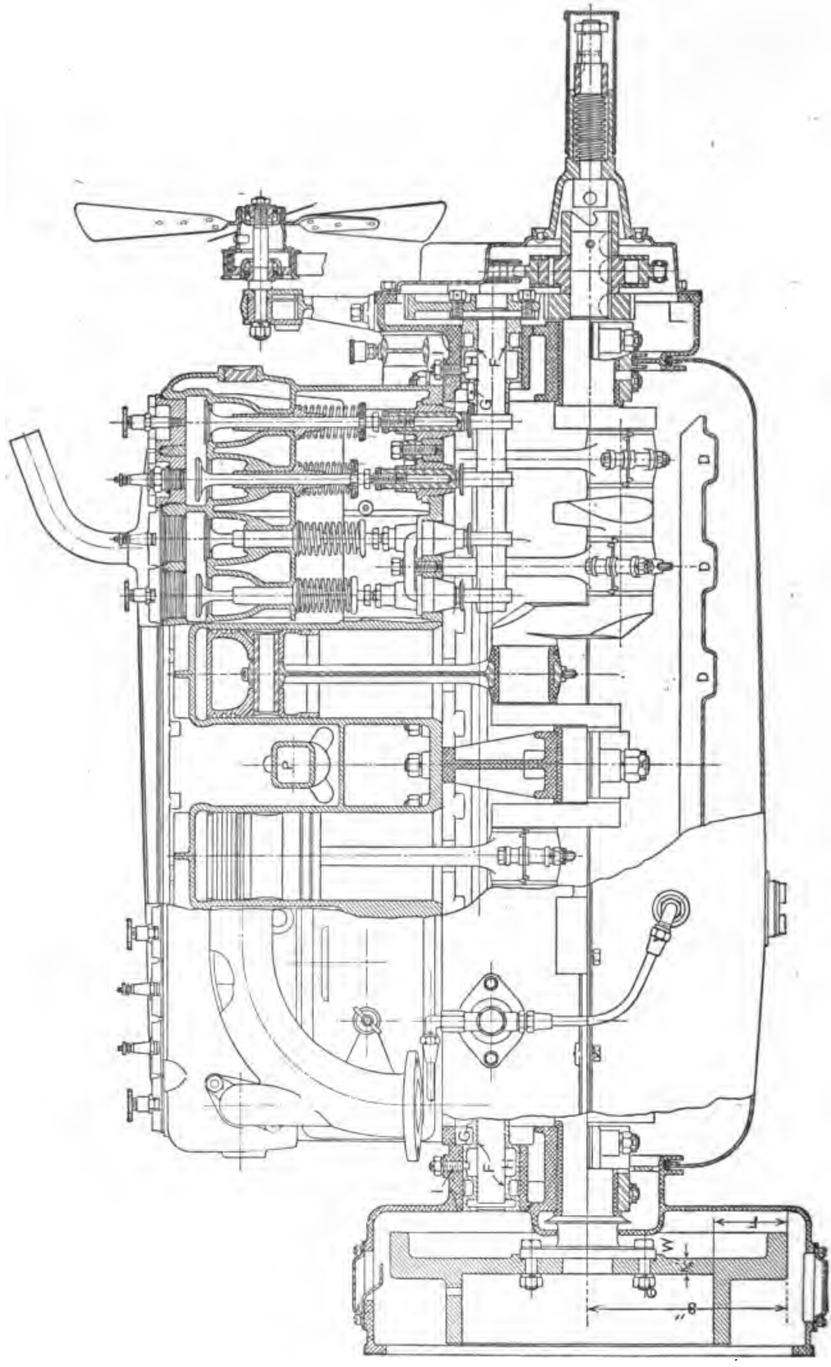


FIG. 35a.—Haynes light-six motor.

is drawn to the profuse amount of water near the valves, and on the top of the combustion chamber. As seen the head separates

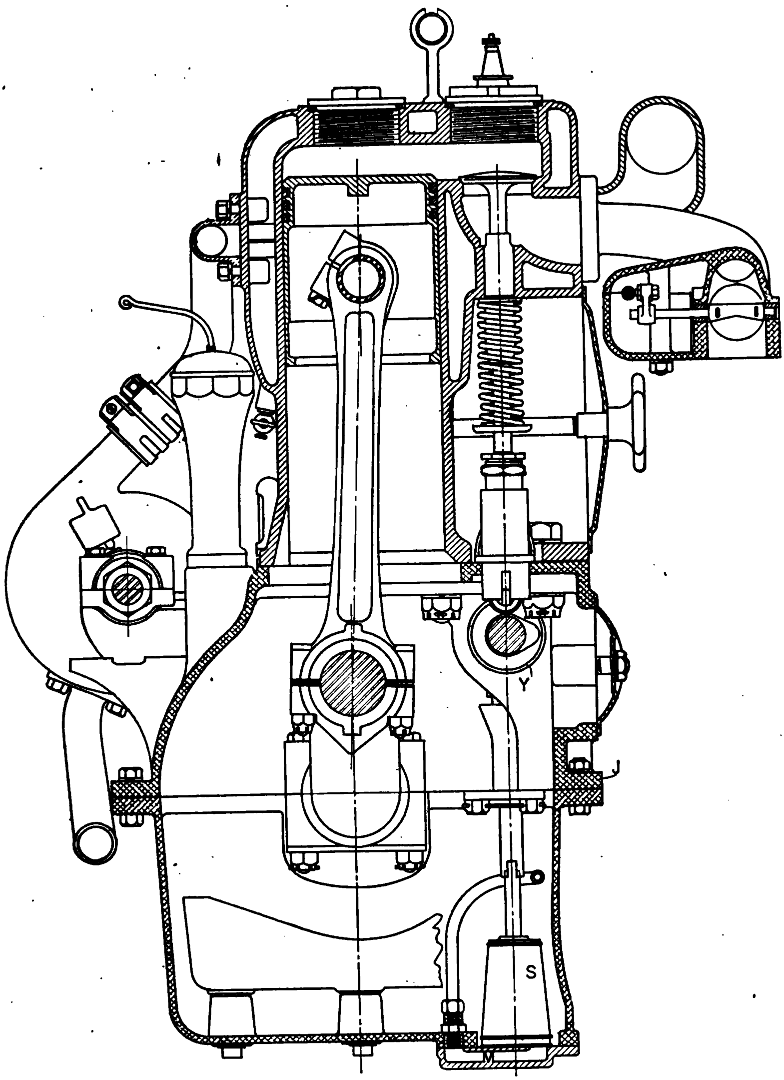


FIG. 36.—Waukesha motor. 4¾ × 6¼.

from the cylinder at "*J*" which is in a plane with the top of the piston, when the latter is in its highest position. By making the joint here rather than higher up, the entire combustion chamber

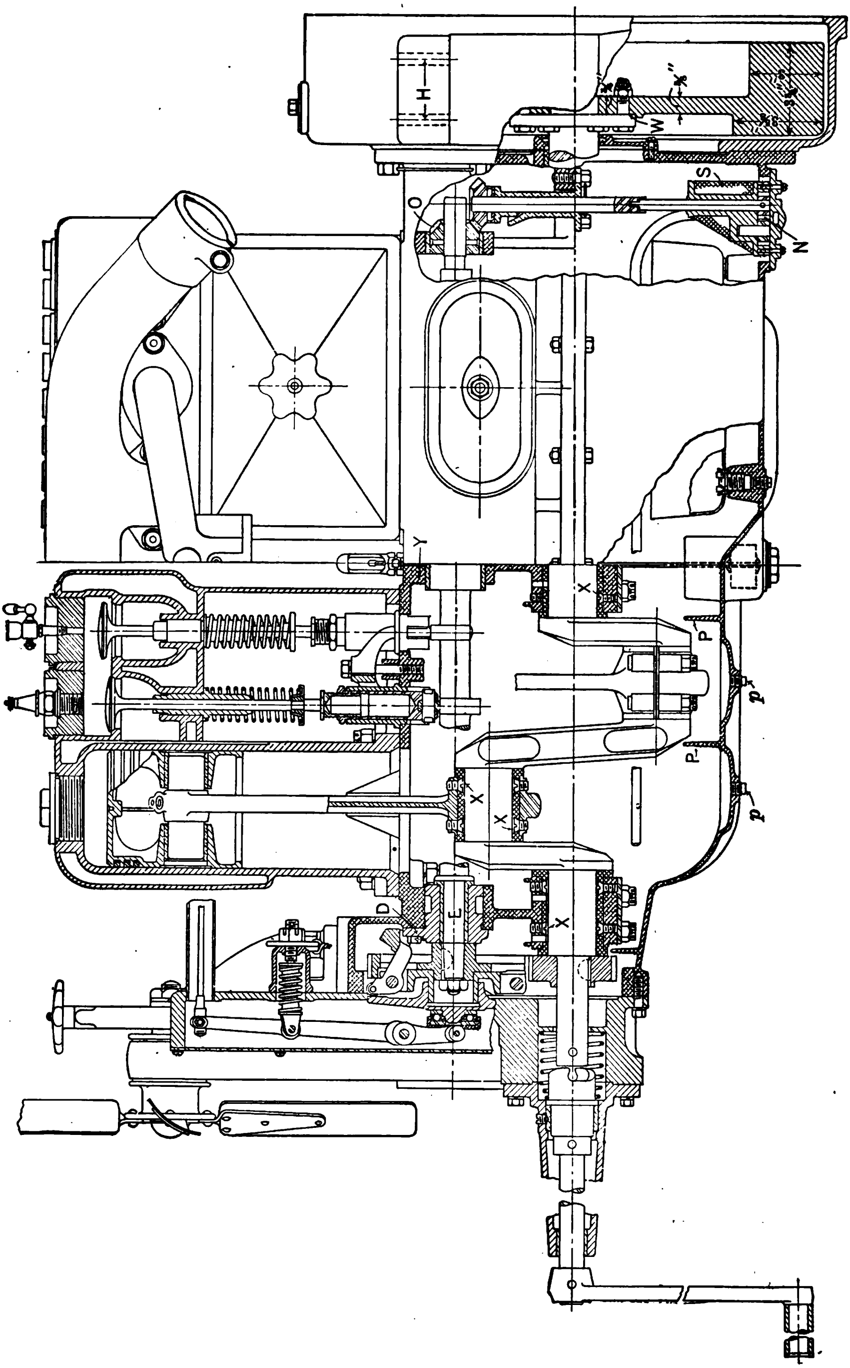


FIG. 36a.—Waukesha motor.

is contained in the head and can more easily be machined. It is also advantageous for scraping the carbon, as the piston can be reached without difficulty while if the joint were higher this

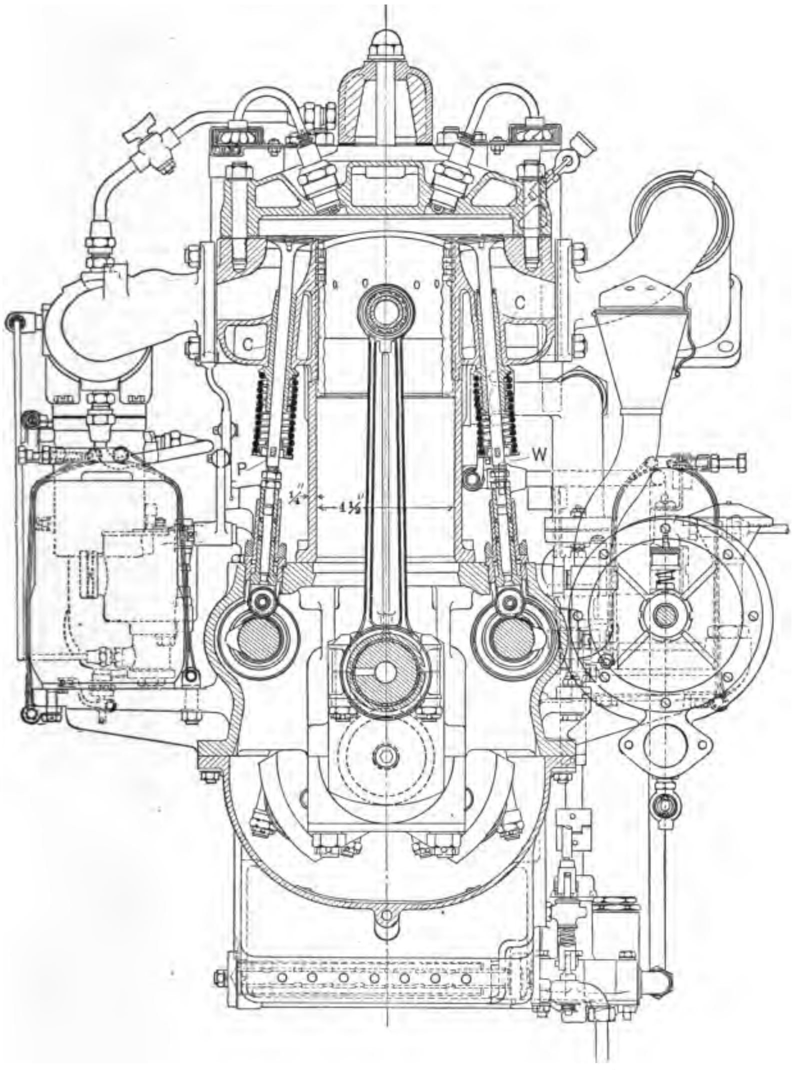


FIG. 37.—Pierce Arrow six-cylinder, 24-valve engine. $4\frac{1}{2}'' \times 5\frac{1}{2}''$.

could not be done so easily. At "*T*" note the extra thickness of the cylinder metal. As previously mentioned, where the

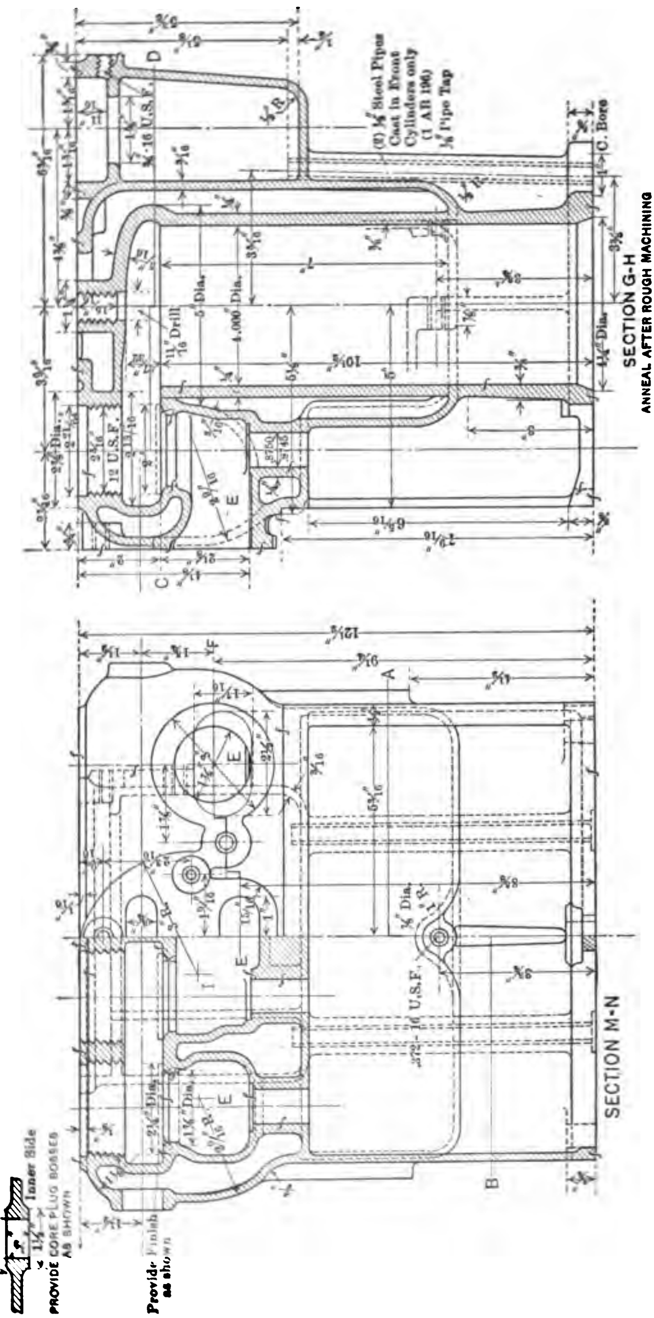


FIG. 38 AND 38a.—Cylinders of Mack engine 4 × 5".

cylinders are fastened to the crankcase the stress is much more severe.

Figures 35 and 35*a* show an end section and a side elevation (partly in section) of the Haynes Light Six Motor. Note the hot-air intake pipe "*h*" which is taken from the exhaust manifold and is carried across between the cylinders to the carburetor, whence the intake manifold "*P*" returns again to the valve side of the cylinder. Figures 36 and 36*a* show the latest design of the Waukesha Motor. Figure 37 shows the Pierce Arrow six cylinder, 24-valve, engine which has just been brought out, the cylinder being of 4½ bore by 5½ inch stroke (this engine will

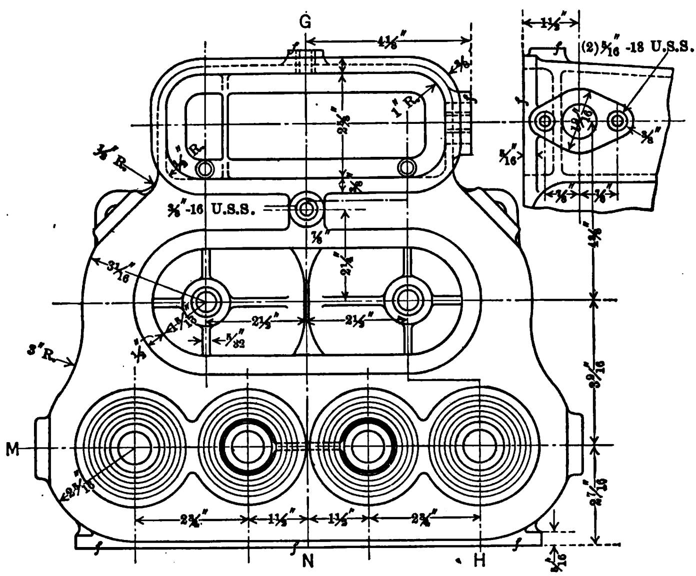


FIG. 38*b*.—Mack engine plan view.

be more fully discussed in subsequent chapters). Figures 38, 38*a* and 38*b* show working drawings of the Mack Engine cylinder manufactured by the International Motor Co. This is a four-cylinder engine with the cylinders cast in pairs; one pair being shown in the drawing. Note with what care the water jacket has been designed to carry the water as closely as possible to all the surfaces to be cooled. The oil tank through which oil is poured into the crankcase, by pipes as shown, is made to form a part of the cylinder casting. This is one of the high-grade

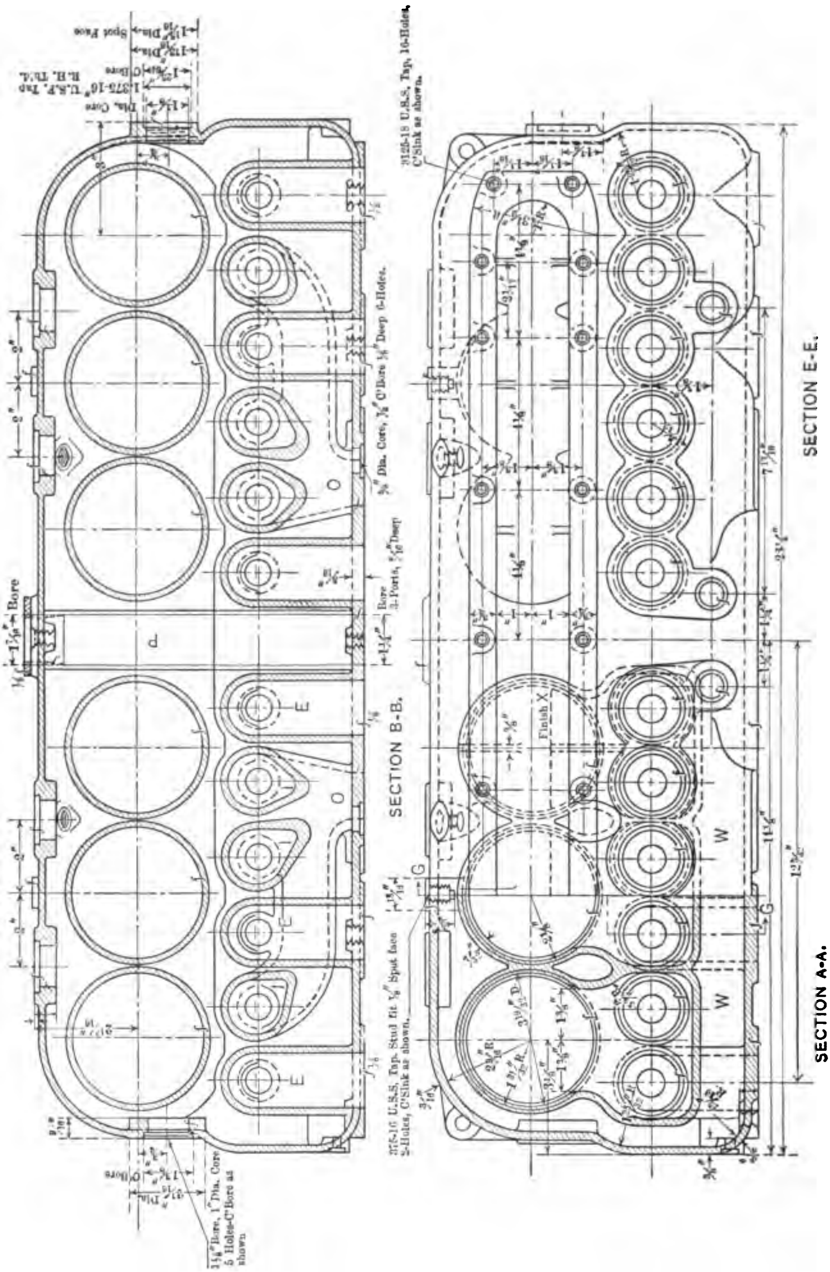


FIG. 39 AND 39a.—Continental Model 9N cylinder casting $3\frac{1}{2}'' \times 5\frac{1}{4}''$.

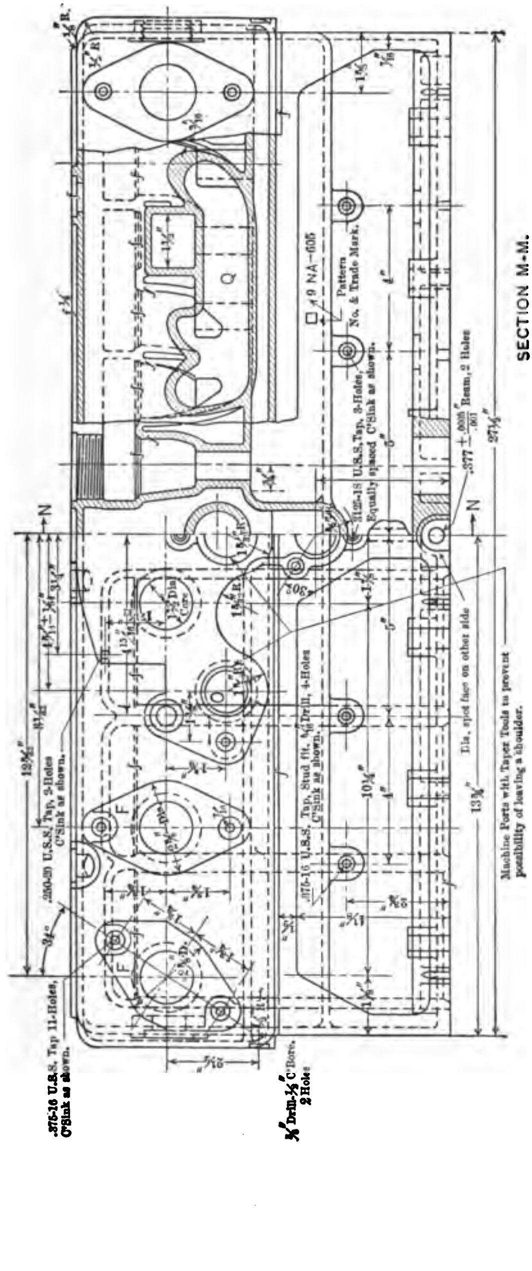


FIG. 39b.—Continental Model 9N $3\frac{1}{2}'' \times 5\frac{1}{4}''$.

Truck engines which has given great satisfaction for several years.

Figures 39, 39*a*, *b*, *c*, *d* and *e*, are details of the cylinder castings of the Continental Model 9N, automobile motor (for passenger

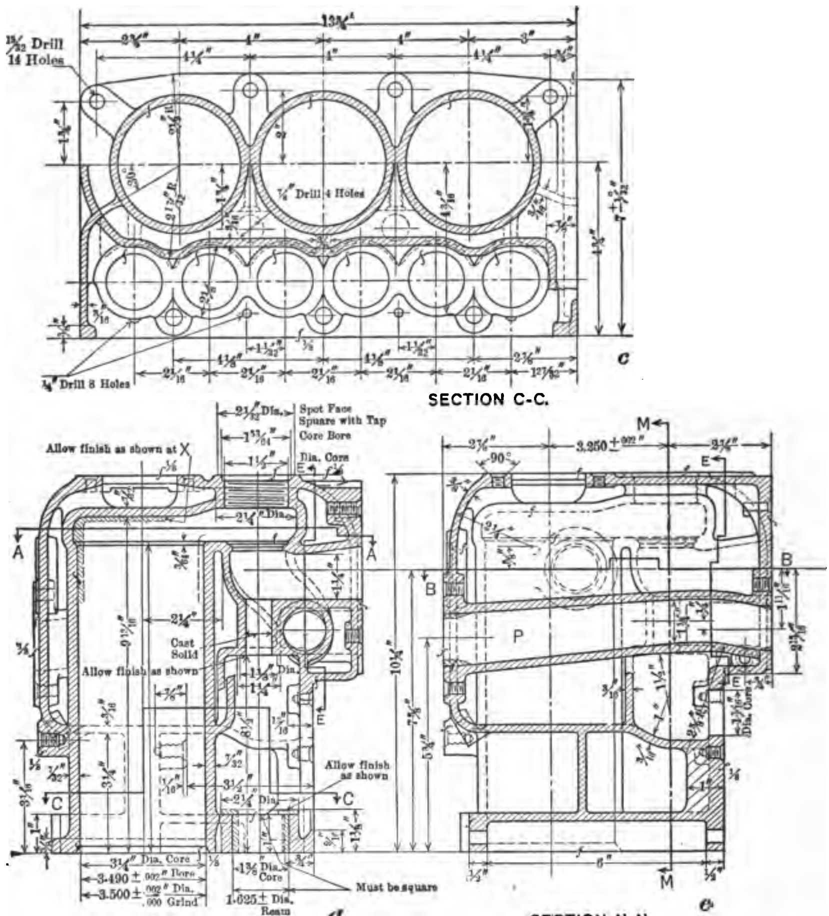


FIG. 39*c*, *d* AND *e*.—Continental Model 9N.

cars). The six cylinders are cast in one block from a special grade of iron. In this motor the charge first enters the cylinder casting from the side opposite the valves, is carried to the valve side, where it is distributed to the intake valves. Unlike the construction shown in Fig. 32, in the Continental motor, the charge is

conducted straight across the cylinder block and into the intake manifold, whence it enters inlet ports *O* (Fig. 39) in each half of the casting. From here it is distributed (in the cylinder block) to the three inlet valves *I* in each half of the block. *Q* in Figs. 39*b* and *d* shows the passage connecting the end cylinders with the inlet ports.

Note the depth of the water jacket *w* around the valves in Fig. 39*a*. The exhaust is carried straight out from each cylinder to the exhaust manifold flanges.

A careful study of these drawings is recommended to all students as this is a late model of one of the most successful engine manufacturers.

CHAPTER VI

VALVES

LOCATION OF VALVES, VALVE DIMENSION, AND VALVE MECHANISM

The different locations of valves greatly affect the design of an engine as well as its appearance and compactness.

The form most commonly used to-day and which is one of the most simple is that shown in Fig. 40*a*. In this case, the valves are arranged side by side in a pocket on one side of the cylinder. Each valve may be taken out through an aperture above it, which is ordinarily kept closed by a valve cap.

In a former chapter we spoke of the importance, theoretically, of keeping the surface area exposed to the hot gases, as small

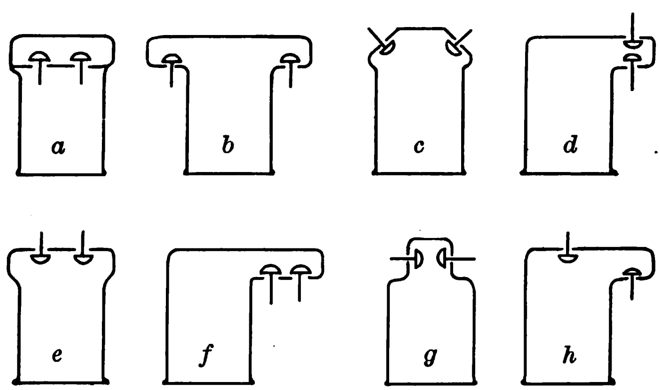


FIG. 40.

as possible, and we mentioned the fact that the largest volume within the smallest surface area of metal is contained in a sphere.

The type shown in Fig. 40*a* (usually called the L head) is not the most efficient from this point of view, as a valve pocket (which is the over-hanging portion of the cylinder which receives the valves) has a large surface of cylinder wall exposed to the hot gases. In the type shown in this Fig., only one cam shaft is necessary to operate the valves (as both the inlet and the exhaust are side by side on one side of the engine), whereas two are necessary

in the valve arrangement shown in Fig. 40*b*. The first-mentioned type is the most common and is used by the largest number of automobiles manufactured, even though the thermal efficiency (theoretically) in this arrangement is not as high as in that of Fig. 40*c*, which is the nearest form to the spherical cylinder head, with the ordinary type of engine in use.

The type shown in *b* (T head) has the largest wall area exposed, and a lower thermal efficiency is therefore expected; on the other hand, it enables larger valves to be used than in the arrangement shown in *a*, and will counteract these losses by an increased volumetric efficiency. It might be stated that some of the best known automobile manufacturers have for years used the T-head motor.

Other types of cylinder heads are shown in Figs. 40*d*, *e*, *f*, *g*, and *h*.

It has been found from practice that when the valves are located in the head, a larger amount of power is obtainable from the engine, which shows that, everything else being equal, engines without pockets have a higher thermal efficiency. In point of efficiency, *e* would be the highest next to that shown in *c*.

In the earlier day of the automobile industry when automatic valves were used for the inlet valves (that is, where the inlet valve was not operated by a cam but simply by the suction or the vacuum in the cylinder working against a spring which ordinarily kept the valve closed) the type shown in Fig. 40*d* was used to a great extent. The valve above is the automatic inlet valve.

The automatic inlet is not used any longer with automobile engines as it could not be timed accurately; the spring would close the valve before the end of the suction stroke when there was still a small vacuum in the cylinder, but not sufficient to overcome the spring. With mechanically operated valves, *i.e.*, where the cams are pressing against the valve stems, or against rods or levers which lift the valve stems, the valves can be opened or closed whenever desired.

The disadvantages of the arrangement in Fig. 40*f* is apparent. It means a still larger or longer valve pocket than that in Fig. 40*a* or *d*, and thus a larger cylinder head.

At the present day, next to the L-head cylinder the I-head, *i.e.*, where the valves are in the head, Fig. 40*e*, is the most commonly used. Approximately 60 per cent. of the various models use

the L head; over 20 per cent. the I-head; about 6 per cent. the T-head cylinders, and about 4 per cent. use the Knight Sleeve Valve engine.

Another advantage of the I-head cylinder is that the whole combustion chamber can be machined, and therefore, in multi-cylinder engines the compression chamber can be made absolutely alike in volume. By having the combustion chamber polished, it is found that the carbon does not adhere to it so readily, therefore, it needs less scraping of carbon. With the I-head cylinder, however, the valve operating mechanism is not so simple.

As mentioned before, the type shown in Fig. 40*a* requires only one camshaft, since the valves are situated side by side, in a single pocket, and therefore only one set of camshaft driving gears are necessary. The camshaft is at one side of the crank-case, directly below the valves and actuates them by direct thrust upward.

When the valves are located on opposite sides as in Fig. 40*b*, two cam shafts are necessary; when they are located in the head of the cylinder a more complicated valve mechanism is required and the larger number of joints or levers give rise to more play in the joints. In some instances, the camshaft is located on the top of the cylinder, in which case the valves can be very easily operated without complicated valve mechanism but the means for driving the cam shaft are not quite as simple. Another important consideration in connection with valve location is the cooling of the valve seats. Unless provision is made for sufficient cooling, especially around the exhaust valve seat, it will become over-heated and the abrasive action of the hot gases will necessitate a frequent regrinding of the valve. When the cylinder head is removable, easy access is had to the valves, whereas when the cylinder heads are cast integral with the cylinder, in the types shown in Figs. 40*c*, *d*, *e*, *g*, *h*, it is necessary to use valve cages which contain the valve seats, and into which the valves are fitted. (Unless this were done, it would be necessary to remove the cylinders to reach the valves.) These valve cages are fastened to the cylinder through holes or apertures. It has been found when valve cages are employed the cooling is not so effective on account of the intervening joint; a joint, as is well known, impedes the transfer of heat from one body to another.

VALVE MECHANISM

In four-cycle engines the valves governing the admission of the explosive mixture to the cylinder and the expulsion of the burned gas from it are called the inlet valve and the exhaust valve respectively. Some years ago as mentioned previously, the inlet valve was automatic in action; as the adjustment of gasoline engines became more and more exact, with the increase of speed, it was necessary to make the operation take place independent of the piston suction. For this reason, the inlet valve, as well as the exhaust valve, are operated by mechanical means, at a given distance of the piston stroke.

In four-cycle engines, one explosion takes place every two revolutions of the crankshaft. Therefore, to operate the exhaust valve and the inlet valve, a second shaft, called the camshaft, has to be employed, rotating at one-half the number of revolutions of the crankshaft, *i.e.*, it revolves once for every two revolutions of the crankshaft.

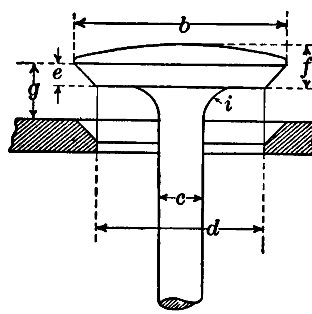


FIG. 41.

In almost all automobile engines the valves are of the mushroom or poppet-valve type; viz., a disk with a beveled surface (Fig. 41) fits into a circular opening which leads into the combustion chamber. In some cases, instead of a beveled surface, a flat-seated disk is used. A stem is attached to the disk projecting downward and resting on the top of a rod in such a position that its lower end is thrust up by a cam on the camshaft, and thrust down again by a strong spring as the camshaft revolves. Every time the rod is lifted by the cam, it pushes the valve up, and holds it open until the cam passes under it, and allows it to drop again into its seat where it is held firmly by means of the spring. At every revolution of the camshaft this operation is repeated, and as it revolves at one-half the number of revolutions of the engine, the inlet and exhaust valves are closed once for every two revolutions of the crankshaft, and reduction gears are employed to reduce the speed of the camshaft.

To overcome the noise of ordinary spur gears, helical gears are used in about 80 per cent. of the different models on the market.

In the helical gears, the teeth, instead of running parallel with the axis of the gear, form a helix, making an angle of 30° with the axis. The helical gear transmits the power more uniformly, as the entire tooth does not come into, nor out of engagement at the same time. Helical gears, however, cause an end thrust, which must be provided for.

The ordinary spur gear, which a few years ago was almost universally used, is not now frequently employed in automobile engines, while the silent chain drive is used in about 18 per cent. of the different models of cars.

VALVE DIMENSIONS

When designing valves the first consideration is the valve area, as we have seen from previous chapters that the efficiency of the engine largely depends upon the size of the valves. The volume of gas swept by the piston has to pass through the valve openings, and the larger the valves the lower the gas speed through them. We may write cylinder area $\times$ piston speed $=$ valve area $\times$ gas velocity. If D is the cylinder diameter, and A the area of the valve, then $.7854D^2 \times$ piston speed $= A \times$ gas speed, and $A = \frac{.7854D^2 \times \text{Piston speed}}{\text{Gas velocity}}$. In practice it has been found the maximum horsepower of an engine is developed when the gas velocity through the valves is about 14,000 ft. per minute. Ordinarily, at the usual speed of the engine, the velocity of the gas will be considerably lower. Knowing, however, the speed of the gas at the maximum horsepower, it is possible to determine the valve dimensions for any piston speed. For instance if it is predetermined that the engine should give its maximum power at a piston speed of 1500 ft. per minute, the area A, of the valve is found from the formula,

$$A = \frac{.7854 \times D^2 \times 1500}{14{,}000}.$$

In the Type B engine previously referred to, we have seen that the bore and stroke decided upon was 4¾ inches $\times$ 6 inches. The speed of the maximum power was intended to be 1500 r.p.m. (the stroke being 6″, the piston speed is equal to the number of r.p.m.)

Our formula in this case is $A = \frac{.7854 \times (4\frac{3}{4})^2 \times 1500}{14{,}000} =$ 1.9 square inches.

In practice in order to have the valve area of sufficient size, additional allowance must be made for the valve stem, and to obtain the full benefit of the area, the lift should be one-fourth the diameter of the valve. This is as a rule too high a lift for quiet running of high-speed engines, and for this reason, the lift is made smaller and the area larger. The formula recommended for determining the valve area and valve lift, and which will give results corresponding with modern practice is to substitute for the speed of the gas through the valves the value 8180. In this case, our formula becomes $A = \frac{.7854 \times (4\frac{3}{4})^2 \times 1500}{8180} = 3.245$ square inches.

The diameter d of the valve will be, $d = \sqrt{\frac{3.245}{.7854}} = 2.035$ $= 2\frac{1}{32}$ inches diameter approximately.

For lift g of valve with the area as found above, when the valve diameter is $1\frac{1}{4}$ inch, the lift is $\frac{9}{32}$ inch, increasing by $\frac{1}{32}$ inch for each $\frac{1}{4}$ increase in valve diameter. To put it into a formula we may write lift $g = \frac{9}{32} + (d - 1\frac{1}{4})\frac{4}{32} = .281 + (d - 1.25).125$, d being the valve diameter in inches. In our example we have $g = .281 + (2.031 - 1.25).125$, approximately $\frac{3}{8}$ inch.

In the type B engine, the valve diameter is $2\frac{1}{8}$ inches (slightly more than found above), but the valve lift was reduced to $\frac{11}{32}$ inch.

When the valves are less than $1\frac{1}{4}$ inch in diameter the lift is ordinarily made higher; the lift in such cases may be made $g = .281 + (d - 1.25).15$.

The other dimensions of the valve can be computed from the following formulæ (see Fig. 41):

$$f, \text{ for cast iron valves } = .14d$$

For steel valves, nickel steel and tungsten steel, f varies between $\frac{1}{8}$ and $\frac{3}{16}$ inch for valves of from $1\frac{1}{2}$ inch to $2\frac{1}{2}$ inches in diameter; for larger valves f varies between $\frac{5}{32}$ inch and $\frac{1}{4}$ inch, depending somewhat on the diameter of the valve stem, the radius of the fillet, and whether it is intended for a very high speed engine, where a light valve is advantageous.

The outside diameter b depends on the thickness of the head; it is about $1.14d$ for cast-iron and somewhat smaller for steel valves. The valve stem c is made according to the table on page 68, in order to make use of standard commercial reamers.

The radius of the fillet is preferably made somewhat smaller when the dome of the valve is high (the valve head fairly thick) or when the stem c is large in diameter; when the valve head is flat on the top or when the stem is small comparatively, the radius is best made a trifle larger to have sufficient metal in the corners. It varies usually between $i = \frac{d}{4}$, to $i = \frac{d}{6}$.

We will proceed to check two or three examples of well-known engines which are in the motor vehicle laboratory at Cooper Union, and see how they correspond with the valve diameters above referred to.

Buda engine—4¼ bore by 5½″ stroke. In this case, we have

$$\text{valve area} = \frac{4.25 \times 4.25 \times .7854 \times 1500}{8180} = 2.601 \text{ square inches.}$$

(Taking the piston speed at 1500 feet per minute.) The diameter $d = \sqrt{\frac{2.601}{.7854}} = 1.82$, or approx. $1\frac{13}{16}$ inches.

The Buda engine has a valve diameter of 1⅞″, *i.e.*, slightly larger than that required for a maximum piston speed of 1500 ft. per minute. In practice, according to test curves made on this motor by the Buda Company, it seems that the maximum horsepower is attained at about 1850 revolutions per minute, which corresponds to a piston speed of about 1700, showing that an increase in valve diameter makes possible a higher piston speed at maximum power. The lift in this case should be $.281 + (1.82 - 1.25).125 = .352$, or approx. $\frac{23}{64}$″. This engine has an actual valve lift of ⅜″ to correspond with a valve diameter of 1⅞″.

Let us take another example of a well-known engine—the motor used on the one-ton Mack truck. In this case, bore and stroke are 4″ × 5″; the valve area according to our formula should be $= \frac{4 \times 4 \times .7854 \times 1500}{8180} = 2.3034$; $d = \sqrt{\frac{2.3034}{.7854}} = 1.715$, or almost 1¾″. The lift should be $.281 + (1.75 - 1.25) .125 = .344 = \frac{11}{32}$″. This engine has a valve diameter of 1¾″ and valve lift is $\frac{11}{32}$″.

In the Continental motor, six cylinder, having a 3½ by 5¼″ bore and stroke, the valve area should be

$$\frac{3.5 \times 3.5 \times 1500 \times .7854}{8180} = 1.764.$$

This gives a valve diameter of 1.5 which the engine actually has. The lift should be $.281 + (1.5 - 1.25).125 = .312$, or approx. $\frac{5}{16}''$. It should be remembered in all these cases, we have taken the piston speed at 1500 ft. per minute.

If the maximum power is desired at a higher piston speed, this speed must be substituted for 1500 in our formula.

The values found before for valve areas are also applicable to valve passages (see chapter on manifolds).

The valves are sometimes made of two pieces, with the stem riveted into a cast-iron disk. It has been found that cast iron can resist the abrasive action of heat much better than steel or wrought iron. On the other hand, nickel steel and tungsten steel have been found very satisfactory, and in the majority of cases, the valves are drop-forged from one piece. In the Fansteel valve the head is punched from a sheet of steel containing from 17 to 20 per cent. tungsten and 1 per cent. vanadium while the stem is made of cast iron, the two being welded together by a special process. The springs in connection with valves vary approximately between 40 and 80 lb., depending on the speed of the engine, and the size of the valve, also on the shape of the cams as we shall see later.

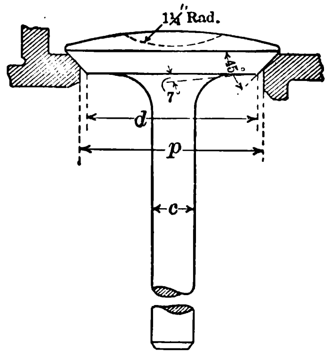


FIG. 42.

Regarding valve dimensions, the society of Automotive Engineers adopted the following standard for valves;

p is the port diameter (smallest diameter), see Fig. 42 and the table following.

d the actual valve diameter (smallest diameter).

c the stem diameter.

The limits for stem diameters have been selected so as to give proper clearances in guides reamed with commercial reamers. The other dimensions not given in the table can be determined from the formulæ given before.

The angle at the bottom of the valve disk leading to the radius is 7° and the slot on the top of the valve which is provided for facilitating the regrinding of valves, is cut with a radius of $1\frac{1}{4}$ inches and is $\frac{3}{32}$ of an inch wide. The two values given under

TABLE II

p	d	c	p	d	c
		0.3090			0.4335
1	$^{31}/_{32}$	0.3100	2	$1^{31}/_{32}$	0.4345
		0.3090			0.4335
$1\frac{1}{8}$	$1^{3}/_{32}$	0.3100	$2\frac{1}{8}$	$2^{3}/_{32}$	0.4345
		0.3090			0.4335
$1\frac{1}{4}$	$1^{7}/_{32}$	0.3100	$2\frac{1}{4}$	$2^{7}/_{32}$	0.4345
		0.3715			0.4955
$1\frac{3}{8}$	$1^{11}/_{32}$	0.3725	$2\frac{3}{8}$	$2^{11}/_{32}$	0.4965
		0.3715			0.4955
$1\frac{1}{2}$	$1^{15}/_{32}$	0.3725	$2\frac{1}{2}$	$2^{15}/_{32}$	0.4965
		0.3715			0.4955
$1\frac{5}{8}$	$1^{19}/_{32}$	0.3725	$2\frac{5}{8}$	$2^{19}/_{32}$	0.4965
		0.3715			0.4955
$1\frac{3}{4}$	$1^{23}/_{32}$	0.3725	$2\frac{3}{4}$	$2^{23}/_{32}$	0.4965
		0.4335			0.4955
$1\frac{7}{8}$	$1^{27}/_{32}$	0.4345	3	$2^{31}/_{32}$	0.4965

c in the table show that the tolerance is .001″, *i.e.*, the stems must not vary more than one one-thousandth inch. One of the latest tendencies, is to use four valves in each cylinder (see Fig. 37) which in this design, for instance, makes possible a larger valve area to be used with smaller valve pockets; this, as we have seen, should give a higher thermal efficiency. It should also be conducive to a more silent valve operation as the valve springs are considerably weaker.

VALVE GUIDES

The valves have to be perfectly seated on their seats to be pressure tight, hence they must be guided accurately. In other words the valve stems must move exactly at right angles to the valve seats. To accomplish this the valve stems are run in bearings called valve guides.

Formerly these guides formed a part of the cylinder casting, but at the present time separate guides are as a rule inserted.

The great advantage of the separate guide is that when it becomes worn it is only required to renew it—a comparatively inexpensive matter—while if guides are dispensed with, in the event of wear, it is more difficult to remedy this defect and more costly.

Various methods are used to hold the guides in place; the most

common is to drive them into the cylinder casting either from above or from below. If forced into position from below, the tension of the valve spring may be used to keep it in place.

Figure 43 shows the valve guide used on the International Motor Co.'s engine, before referred to, while Fig. 44 is the guide used on the latest Waukesha engine having a bore and stroke of $4\frac{3}{4}$ inches $\times$ $6\frac{3}{4}$ inches (see also Fig. 37). In these figures, *C*

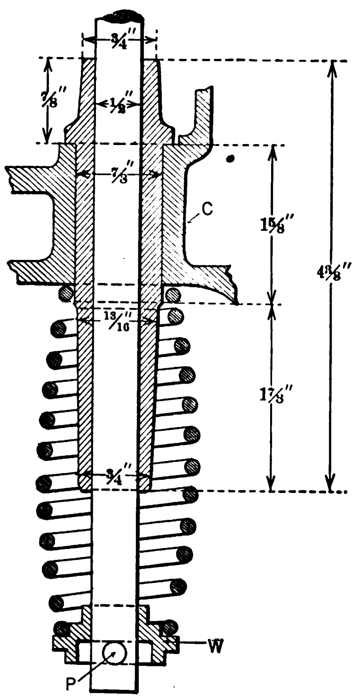


FIG. 43.—Valve guide of Mack engine.

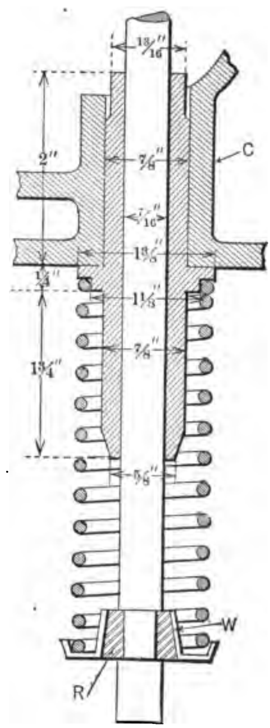


FIG. 44.—Valve guide of Waukesha motor.

represents the cylinder casting into which the guides are fitted. The dimensions are given and these two examples may be said to represent typical designs of valve guides. These figures also show the methods used for holding the valve springs in place by the valve stems. In Fig. 43, a pin *P* passes through the valve stem which holds the valve spring collar *w* in position, this in turn supports the spring.

In Fig. 44 the steel ring *R* is split in two and placed in position,

in the recess of the valve stem, when the spring with spring collar w is raised. When w descends over the ring R, the latter is firmly held on the stem and it also holds the spring in its place. Instead of making the corners of R square, as shown, where they fit into the valve stem, it is well to round them off and to provide a fillet in the corners of the recess; breakage of the stem sometimes occurs when the recess is made with square corners.

VALVE SEATS AND VALVE CAGES

In most engines constructed at the present day, not using over-head valves, the valve seats are made integral with the cylinders. In over-head valves the valves are held in a cage which is complete with the valve and the cap holding the valve cage. With detachable cylinder heads of course the cage can be eliminated. The disadvantage of making the valve seats integral with the cylinders is that every time it becomes necessary to regrind the valves the valve seat is broadened, and eventually the valve seat may become too large for a given valve. Where a cage is used, the cage would simply have to be replaced in such event. The integral seat, however, is more efficient in cooling as there are no joints obstructing the conductivity of the metal.

Different methods are used for holding the cage in place, sometimes a yoke is used, other times bolts are employed to screw it down or else a separate cap is made use of, which is screwed into the cylinder on the top of the cage.

PUSH RODS

In most engines the valves are raised by the action of the camshaft, but instead of having the cams press against the valve stems directly, another rod, called the push rod, is placed between them, see Figs. 46, 47, 48 and 50. There are a number of reasons why a push rod is necessary, instead of applying the cams directly against the valve stem, as will be mentioned later. The upper part of the push rod is usually made solid for very small engines, and tubular for other sizes. A guide, called the push rod guide, is required to keep the push rod in position; it is ordinarily made of cast-iron or hard brass and is fastened to the crankcase.

The lower part of the push rod is either made flat (to be used with certain designs of cams, see Chapter VII) or is provided with a roller as shown in the illustrations. When the valves are located in pockets a direct push rod is employed, that is to say

the push rod comes in contact with the valve stem. When valves are placed in the cylinder head, as in I-head cylinders, the camshaft might be placed on the top of the cylinders; if it is placed near the crank shaft, as is usually the case, push rods, tappet rods and rocker arms have to be employed. The principle of all push rods is a rod sliding in a bearing and lifted by a cam. There is a certain amount of play or clearance between the end of the push rod and the valve stem to make allowance for the

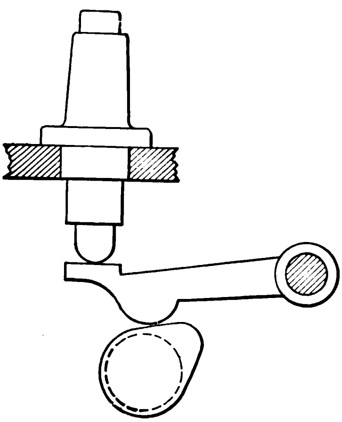


FIG. 45.

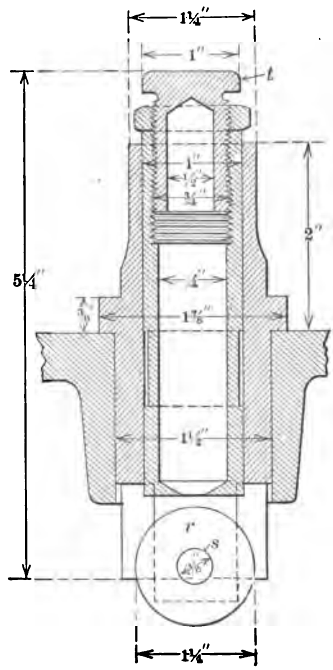


FIG. 46.—Push rod of Mack engine.

expansion of the valve stem, when it becomes hot. For an inlet valve the clearance is usually about .003 or .004 inch, whereas for exhaust valves it is .004 to .006 inch. In truck motors, where the engine speed is not as high as in high-speed touring-car motors the clearance (tappet clearance) is frequently made greater than the dimension given. Where the valve stems are long, they will naturally expand more when they become heated, and they will require a larger clearance.

The upper end of the push rod is usually made adjustable by having an adjusting lock nut of some kind, on its end, in order

to be able to adjust the clearance between the push rod and the valve stem when the valve is seated. Some manufacturers place the push rod not directly over the center of the cam shaft but a little out of center, in order to have less side thrust. There always is more or less side thrusts in connection with the rod when the valve is opening and closing. This side thrust causes the push rod guide to wear, so that after it has been running for a long time there is an amount of lost motion or play between the

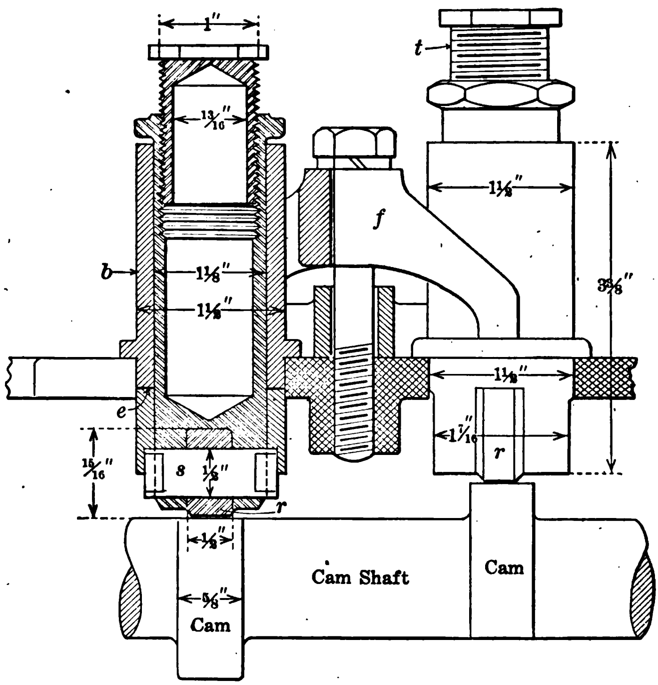


FIG. 47.—Push rod of the Class B and the Waukesha motors.

bearing and the rod. This play or wear causes a great deal of noise when the valve is opening and closing and it may make a difference as regards timing to which the valves were originally set.

In motors where the valves are in the cylinder head, two systems are in vogue to operate the valves. The system most frequently used is a long tappet rod above the push rod, working a sort of a rocker arm, which opens the valve when the cam raises the push rod. The other type is to place the cam shaft on the top of the cylinder, the cam shaft being driven by beveled

gears or chains. In order to minimize the side thrust of the push rod, the guide should be placed as closely as possible to the cam. Sometimes the push rod is so arranged that under normal conditions by means of a spring it is held in contact with the cam. Usually, however, no spring is used in connection with it. In order to overcome the side thrust of the push rods, in some engines, small levers, called valve lifter levers (see Fig. 45), are

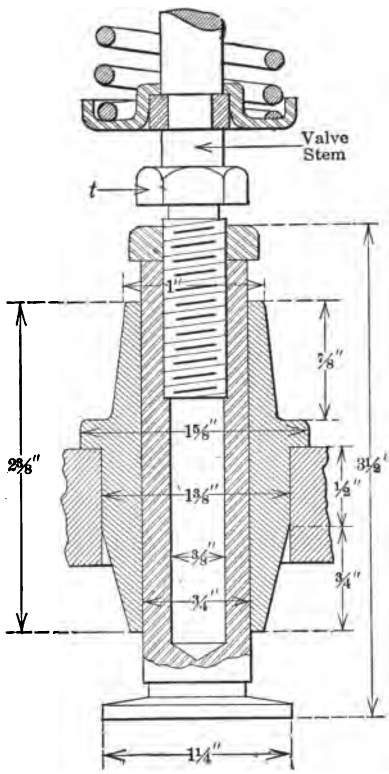


Fig. 48.—Push rod and guide of Haynes light-six motor

employed between the cam and push rod, the lever being pivoted to some convenient point while the other end may have a roller on the portion touching the cam. (The portion of the push rod touching the cam is the cam follower.) As a rule no valve lifter levers are used and the bottom of the push rod contains a roller. Very often, however, no roller is employed, the end of the push rod touching the cam being shaped like a mushroom. This is called the mushroom type of cam follower (see Fig. 48).

Figure 46 is the push rod used on the International Motor Co.'s engine, while Fig. 47 represents the push rod used on the latest Waukesha engine referred to; it may be said that the latter is practically the same as that used on the Type *B*, Quartermaster engine. Both these examples show at *r* the roller type of cam follower.

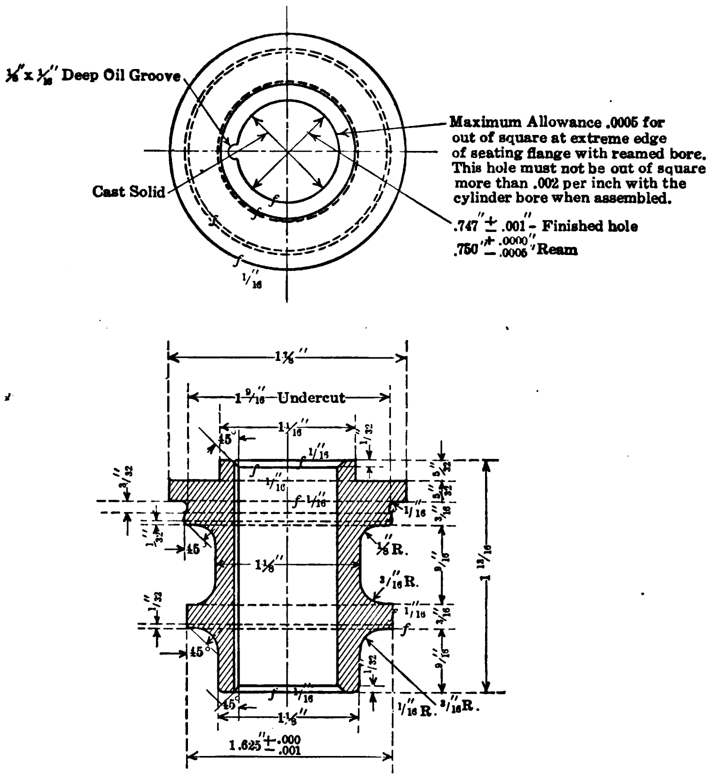


FIG. 49.—Push rod guide (cast iron) of Continental engine.

The rollers at the bottom of the push rods vary in size between ⅞ inch and 1¼ inches; 1 inch is about the average size used, while the shaft "*s*" of the roller is as a rule made from one-third to one-half the diameter of the roller. The push-rod guide *b* is slotted at the bottom up to *e*, to permit *s* (Fig. 47) to slide upward and prevent the roller form turning.

Figure 48 shows a push rod with mushroom type of cam follower as used on the Haynes Light Six engine. The dimensions are given and need no further explanation. Figures 49 and 50

show the cast-iron guide and the mushroom push rod of the Continental motor.

The adjustments for regulating the clearance between valve stem and push rod are made by turning the threaded portion *t* to the right or the left until proper clearance is obtained (see

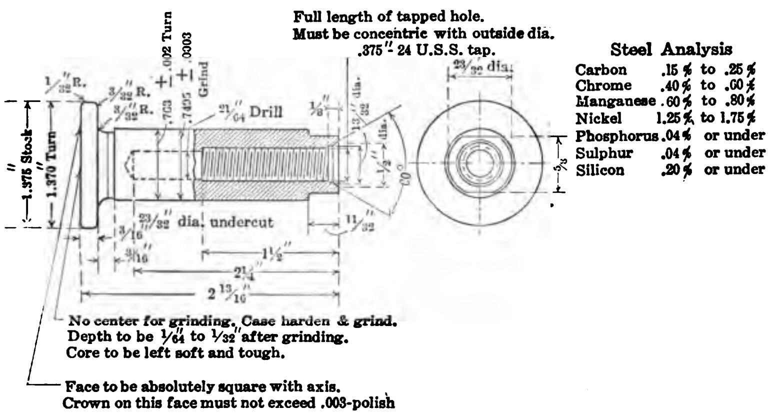


FIG. 50.—Push rod of Continental engine.

Fig. 47). The push rod shown on the right is indicated in its raised position. The forked clamp *f* shows a very common method of holding the push rods in place.

Figure 51 shows a typical valve cap design which is used over the valves in the L-Head or T-Head cylinders; note how closely

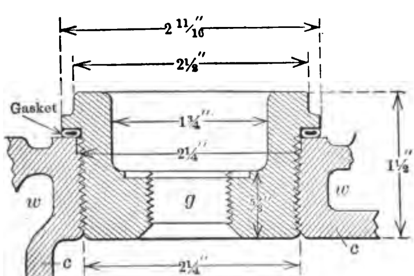


FIG. 51.—Velve cap of Mack engine.

the water is brought to the valve cap to prevent it from overheating. As a rule the spark plug is screwed into the opening *g* of one of the caps while the priming cock is placed into the other cap. Unless detachable cylinder heads are used, it is, of course, necessary to use a cap over each valve in L-Head and T-Head engines.

Figure 52 shows the over-head valve arrangement as used on the Nash six-cylinder engine. Here 1 shows the valve push rod adjusting nut with lock nut 2; 3 is the tappet rod or push rod and 4 the valve rocker arm which revolves around axis 5. As the cam raises the push rod, the valve is opened through the rocker arm.

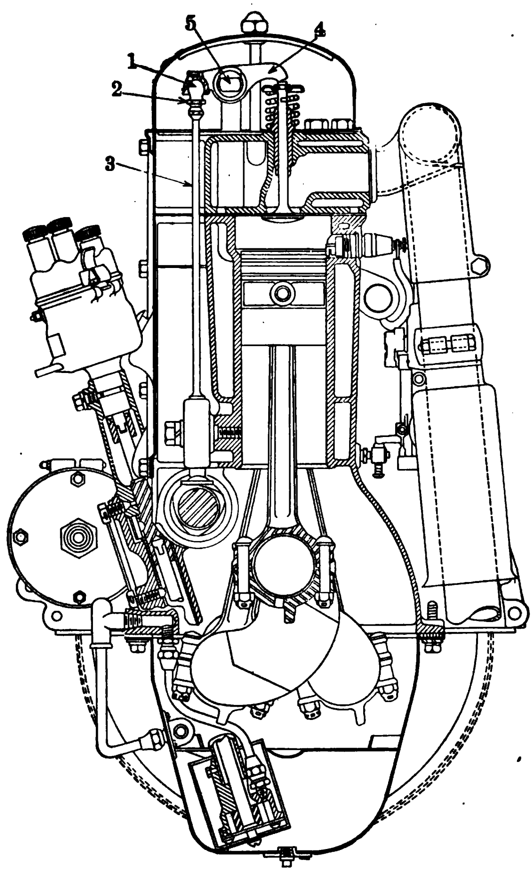


FIG. 52.—Nash 6-cylinder engine.

Figure 53 shows the arrangement of the Reo four-cylinder motor (4⅛-inch bore, 4½-inch stroke) with the inlet valve in the cylinder head and the exhaust valve in a valve pocket. Figure 54 shows the details of the over-head valve mechanism with the push rod. The inlet valve is considerably larger than the exhaust valve (there being sufficient room as it is the only valve in the head),

thus making possible a high volumetric efficiency of the engine. The inlet valve is $2\frac{5}{16}$ inches in diameter and the exhaust $1\frac{3}{4}$ inches. The cylinder head is cast integral with the cylinder casting and no valve cages are employed, therefore it is necessary to detach the cylinder entirely from the crank case and remove the piston from the cylinder, before the inlet valve can be removed. The inlet valve, however, does not require as much attention as exhaust valve since it does not get as hot. To

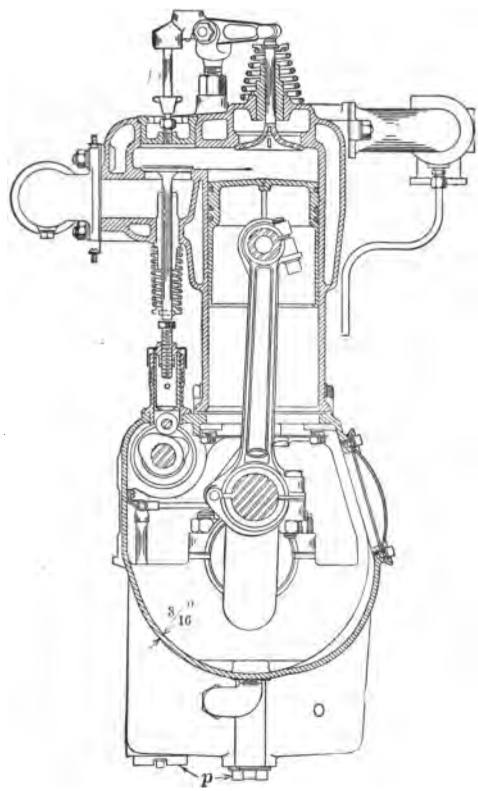


Fig. 53.—Reo 4-cylinder motor. $4\frac{1}{8}'' \times 4\frac{1}{2}''$.

remove the exhaust valve all that is necessary is to remove the valve cap above it.

In Fig. 54, 1 shows the oil retainer for the valve lever; 2 is the seat for upper end of valve push rod; 3 is the upper end of the push rod which is made of steel; 4 is the support for the valve rocker arm. It should be noted that the push rod 5 in its vertical travel will also move slightly horizontally, as the rocker arm

moves on axis 6; for this reason a ball and socket is provided at 7 which permits lateral motion of upper part of rod 5. The roller 8 on the rocker arm is ¾ inch in diameter. 9 is a dust band for the inlet valve lifter.

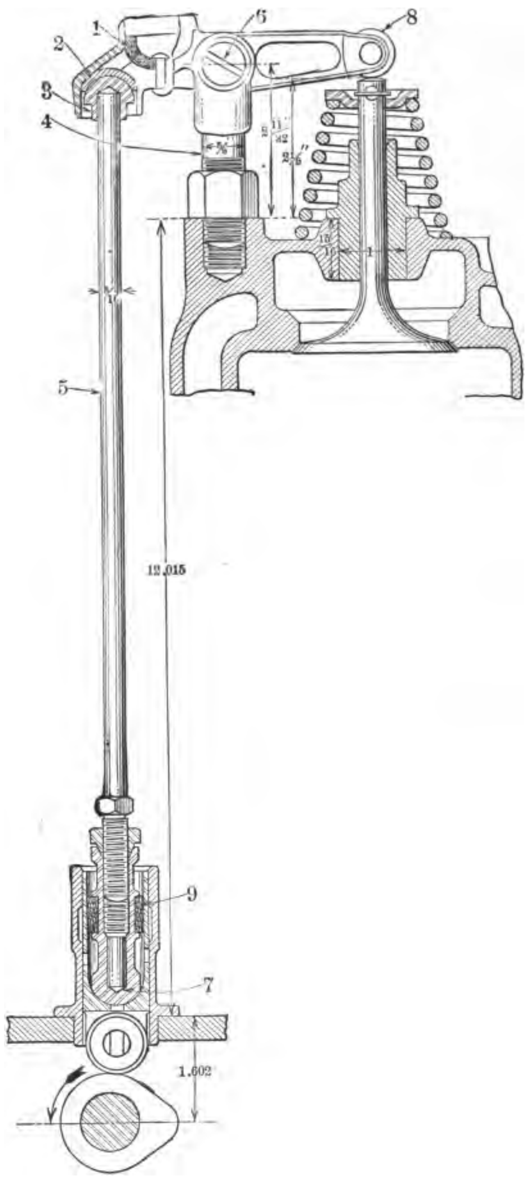


Fig. 54.—Inlet valve mechanism of Reo motor.

CHAPTER VII

VALVE TIMING, CAMS, AND CAMSHAFTS

Correct valve timing is of very great importance, since too early or too late an opening or closing of either the inlet or the exhaust valve will affect the horsepower and the thermal efficiency of the engine. No rule can be laid down as to the exact time when the valve should open or close, for this depends on a number of conditions, and on the general design of the motor. The shape of the cams will also influence the time when valves should begin to open or close. The following, however, may be said to represent the average timing, as determined from a number of models on the market. The inlet valve begins to open at about 10° past upper dead center and closes at 40° past the lower center. The exhaust valve opens at about 47° before lower center and closes at 7° past upper center. Where the motors are run at a slower speed, as for instance in truck engines the timing is somewhat different. For engines running normally at 1000 revolutions per minute the inlet opens at about 5° past top center and closes at 35° past bottom center. The exhaust opens at about 42° before lower center and closes 5° past upper center. On the other hand for very high-speed engines the inlet will open at about 15° past upper center and close at 45° past bottom center. The exhaust will open at about 50° before bottom center and close at 10° past top center. It is interesting in this connection to note the timing of the Type B, Military Truck which we discussed at various times. In this engine the inlet opens at 12° past upper center and closes 35° past lower center. The exhaust opens 45° before lower center and closes 5° past upper center.

When the engine is designed for a certain speed and is timed to have the maximum efficiency at that speed, it will not run so satisfactorily at other speeds nor be as economical. For this reason it is advisable to time the engine so that it should be most economical at the speed at which it is mostly intended to be run. For instance, suppose an engine was timed that the exhaust valve

opens, let us say at 50° before lower dead center, if this engine is run at low speeds the expansion curve will drop to zero before the piston reaches the end of its stroke, thus resulting in a loss of power. At the end of the exhaust stroke the pressure will drop to atmospheric, and at the beginning of the suction stroke, the exhaust being still open, some of the burned gas will be drawn back into the cylinder; at high speeds this of course would not occur. Likewise when the inlet valve closes very late some of the charge will be forced out again.

CAMS AND CAMSHAFTS

The push rod is raised by a part of the camshaft called the cam, see Figs. 46 to 54. Theoretically the best cam would be one which opened and closed the valve instantaneously. In practice this is not feasible as such rapid action would resemble a severe blow, introducing excessive stresses (which means rapid wear) and be very noisy. Therefore the cam must be so shaped as to open and close the valve gradually; the shape of the cam follower also influences the valve timing as will be discussed later.

The cams in use at the present day are: the tangential cam (Fig. 55), the convex flanked (Fig. 56), and the constant acceleration cam (Fig. 57). With the tangential cam and the constant acceleration cam, the roller cam follower is used, as illustrated in Figs. 46, 47, 53 and 54; or just a round or V-type follower like the bottom of the push rod illustrated in Fig. 45. With the convex flank cam the follower shown in Figs. 30, 35, 48 and 56, called the mushroom follower is employed. At the present time the tangential cam with the roller cam follower and the convex cam with the mushroom follower are provided in the greatest number of models on the market. With this cam the valve closes very quickly thus requiring a strong force or pressure on the valve, else the cam follower might not follow the cam fast enough, on account of the speed and the inertia of the driving parts. In practice this pressure is exerted by the valve spring.

The mushroom cam follower (which operates with the convex flanked cam), while simpler in construction than the roller cam, introduces sliding friction. This type of cam and cam follower, during the lift, accelerates very readily (it begins to open the valve very rapidly), about three times as fast as with the tangential cam but it decelerates at a much slower rate (it starts

to close more slowly). For this reason a considerably weaker spring can be employed. With the convex flanked cam the valve begins to close as soon as it has reached its highest lift, for this reason the cam has no flat portion on the top as is sometimes the case with the tangential cam for the exhaust valve.

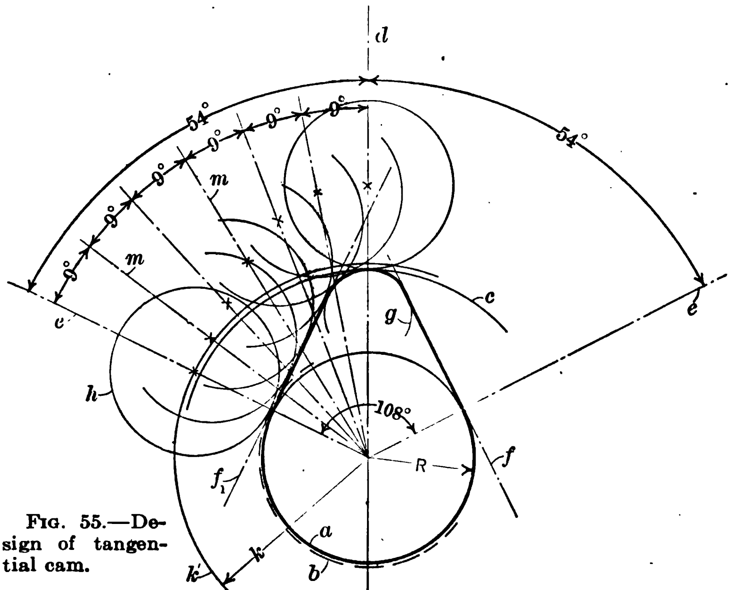


FIG. 55.—Design of tangential cam.

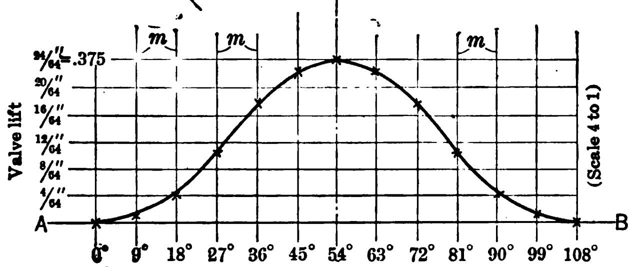


FIG. 55a.—Curve showing valve lift.

(The constant acceleration cam lies about half way between the values of acceleration of the other two.) Having the most rapid acceleration, it (the mushroom type) hits the push rod the hardest and is therefore the most noisy, while the tangential cam being the slowest in acceleration would be the least noisy; the noise however, in all types has been very largely overcome by having the whole push rod mechanism enclosed. For the

reasons before mentioned the tension of spring is greatest with the tangential cam and is smallest with mushroom type, while with the constant acceleration cam the tension of valve spring is between the two, as the acceleration of the valve opening is between the two. As an example let us say in the first type the spring is 65 lb., in the mushroom type it would be about 35 lb., and with the constant acceleration cam about 45 lb.

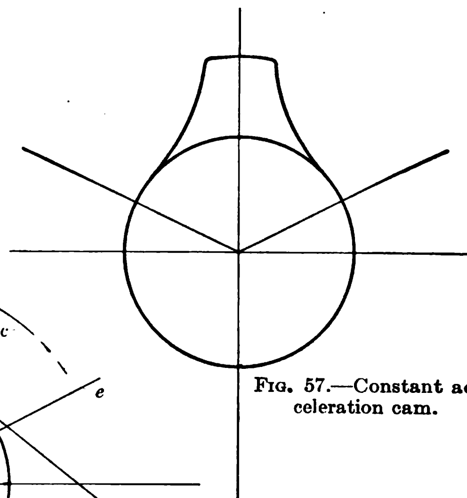


FIG. 57.—Constant acceleration cam.

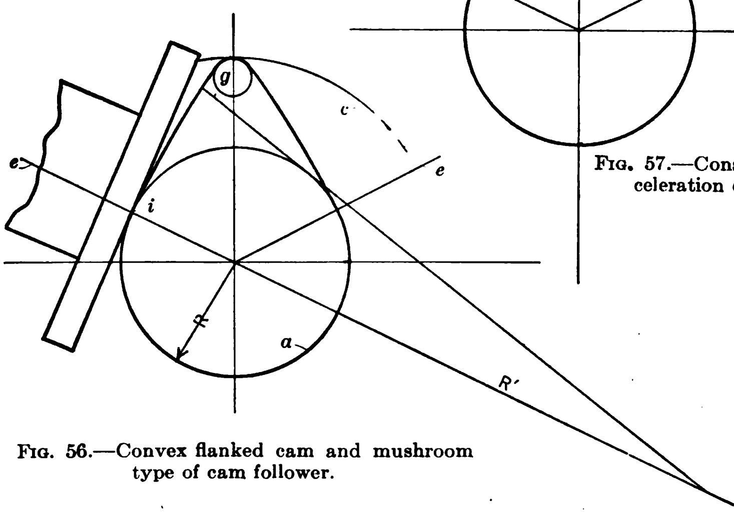


FIG. 56.—Convex flanked cam and mushroom type of cam follower.

The constant acceleration cam is so named as its outline can be made to give the valve a constant acceleration during the first part of its lift and during the first half of its closure, and a constant deceleration on the last half of the lift and during the last half of its closure. In designing the cam, however, both the acceleration and the deceleration can be timed as desired by designing the cam accordingly.

Figure 58 shows the curves of the lift for the three forms of cams. It can be noted that while the total time of opening for the three types of cams is the same the valve will be opened more

rapidly with the mushroom type than with the tangential cam. For instance, at the cam shaft travel of 20° the lift of the push rod (and therefore of the valve), with the mushroom follower will be about .165 inch, while with the tangential cam it will only be .070 inch, showing that at 20° of travel, in the former the valve will have opened almost two and one-half times as far as with the latter.

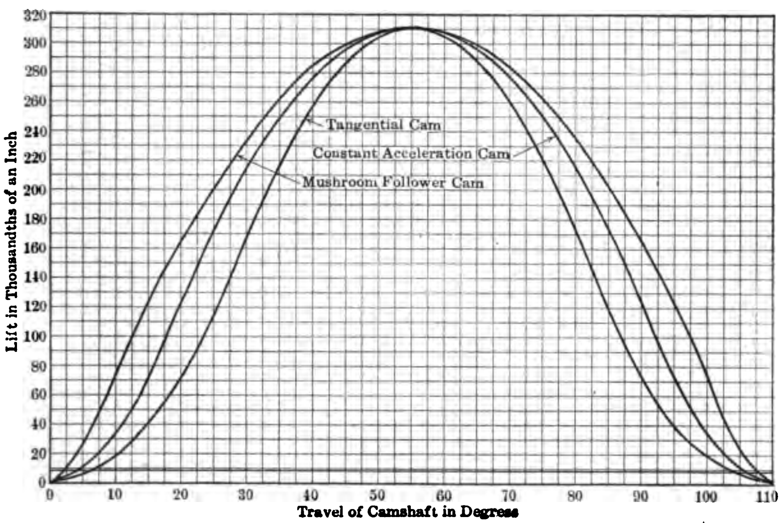


FIG. 58.—Valve lifts with the three types of cams in use.[1]

DESIGN OF CAMS

When laying out or designing cams, we must first know the exact timing of the valves. Having decided that the inlet valve is to open at 4° past top center and close at 40° past bottom center, which means that the inlet valve will be opened during the crank motion of 180° − 4° + 40°, that is to say, = 216°. Since the camshaft revolves at one-half the speed of the crankshaft the angle required will be 216 divided by 2 equals 108°. To design a tangential cam for this valve timing we first lay out a circle *a* (see Fig. 55), whose radius is as a rule from 1/64 to 1/16 inch larger than that of the camshaft to provide a wearing surface and to facilitate the grinding of that surface. Then we draw another circle *b*, which corresponds to the play or clearance between push rod and valve stem. We next draw a portion of a

[1] From Heldt's "The Gasoline Automobile."

circle *c*, which corresponds to the maximum valve lift. Let us assume this lift to be three-eighths of an inch, in which case the radius of circle *c* would be = radius *R* + height of clearance + valve lift. If the radius *R* is three-fourths of an inch, the clearance $\frac{4}{1000}$, the radius of circle *c* will be = .75 + .004 + .375 = 1.129, which is only $\frac{4}{1000}$ more than one and three-eighths of an inch. We then draw a vertical line *d*, which is the center of the cam and lay off the angle of 108°, that is to say, 54° on each side of the center line *d*, the lines *ee*, represent this angle of 108°. Next draw the lines *f* tangent to circle *a*, that is to say, at the periphery of circle *a*, and at right angles to lines *e*. (A tangent is a straight line which touches the circumference at only one point, called the point of tangency or point of contact. A tangent is perpendicular to a radius drawn to the point of tangency). Then draw a small circle *g*, which will touch circle *c* on the top and at the sides of the lines *ff*. The roller *h* must begin to lift when its center reaches line *e*, therefore, the clearance should be taken into consideration as will be shown later. The roller here represents the cam follower at the bottom of the push rod.

If it is desired to find the actual valve opening for each position of the cam, draw a circle *k'* (with radius *k*), representing the height of the center of the cam rollers, from the base line *a*. Then divide the whole angle of valve opening (that is to say 108°, in this case), into a number of equally spaced angles by lines *mm*, and draw circles of the size of the cam roller, touching the cam outline and having their centers on the lines *m*. The distances between the center of the roller position *xx* and circle *k'* represent the actual lift of the valve for the various angles of the camshaft noted. To plot the curve representing the amount of valve opening for every angle of the camshaft, draw a line *AB* (Fig. 55*a*), and divide it into the same angles, denoted by lines *mm*, as in Fig. 55; the total angle being 108°, or whatever the valve period may be. Then mark a small cross at each angle (on lines *mm*) and let the height and lift for each angle be represented by the distance between small cross and line *AB*.

It might be mentioned that if the angle of valve opening is very large, as often required with the exhaust valve, there may be a flat portion on the top of the cam along circle *c*, and the corners simply rounded with a radius of from about $\frac{1}{16}$ inch to $\frac{1}{4}$ inch (depending on the valve lift and the length of flat portion).

Students are recommended to draw the cams and the diagrams to a scale of at least 4 to 1 to obtain greater accuracy.

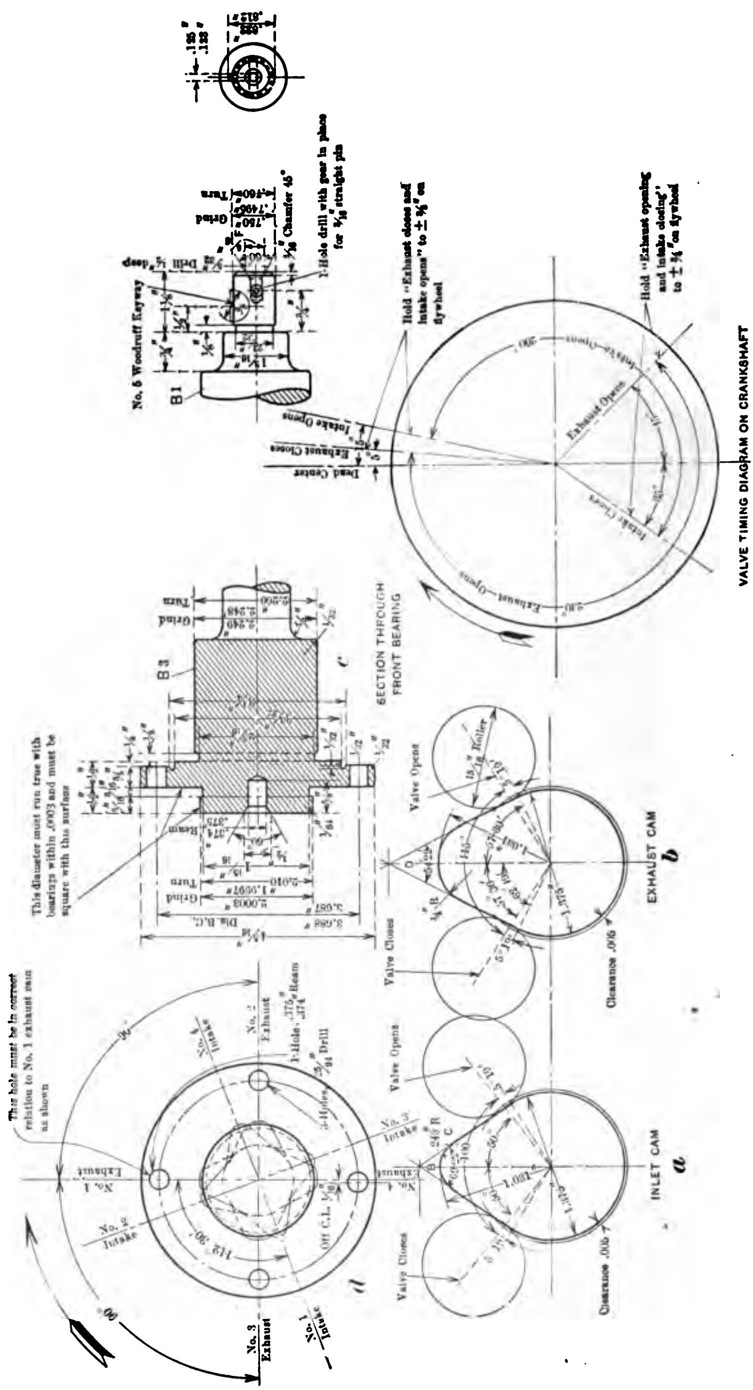
CONVEX FLANKED CAM

To design a convex flanked cam (Fig. 56), as used with the mushroom type of cam follower, proceed as follows: First strike the circles *a* and *c*, and lay out the lines *ee*, representing the angle during which the valves should open and close as in the preceding case. Next draw a small circle *g*, touching at its highest point the arc *c*, which represents the maximum valve lift. The radius of the small circle *g*, depends on the acceleration and deceleration which it is desired to impart to the cam follower. (The larger the circle *g*, the longer will the valve be open near the maximum lift.) Then draw the flank with radius R' tangent to circle *a* at point *i*, and tangent to small circle *g*. With the mushroom type of cam follower and the convex flanked cam, the acceleration of the valve lift is more rapid in the beginning, and the deceleration is slower on closure of the valve. When designing the constant acceleration cam (which is not used so much), the acceleration and deceleration can be predetermined and a curve drawn showing the desired valve lift at every angle, then a number of circles are drawn (to correspond with the amount of lift) at each angle of the camshaft on the cam outline. The operation is reversed from that pursued when drawing a curve with the cam outline first determined, as was done in Figs. 55 and 55*a*.

Figure 59 shows the valve timing diagram of the U. S. A. Class B Military Truck Engine, and Figs. 59*a* and 59*b* show the details of the inlet cam and the exhaust cam respectively; Figs. 59*c*, 59*d*, 59*e*, *f*, and 59*g* show working drawings and details of the camshaft. From these complete working drawings the student is enabled to see all the dimensions that have to be predetermined. This engine has a roller cam follower like that illustrated in Fig. 47.

The diagram (Fig. 59) shows that the intake opens 15° after upper dead center, and closes 35° past lower dead center; therefore, the intake will be open for a crankshaft angle of $- 15 + 180 + 35 = 200°$. On the camshaft this will mean 100°, since it revolves at one-half the number of revolutions of the crankshaft.

On Fig. 59*a* note in addition to 100° on the cam an allowance on each side of 5°19″. This is the allowance found necessary for the clearance and can be determined when drawing the cam



FIGS. 59, 59*a*, *b*, *c*, *d* AND *f*.—Valve timing diagram cams and camshaft of the Class B military truck engine.

outline on an enlarged scale. The clearance, as seen, is made .005 for designing the cam outline. In other words the angle of action and the clearance angle on the cam outline will be 100° + 5°19′ + 5°19′ (but the angle of action of the valve proper will only be 100°); since the cam follower has to be lifted through the clearance as seen from Figs. 59*a* and 59*b*, before it can lift the valve.

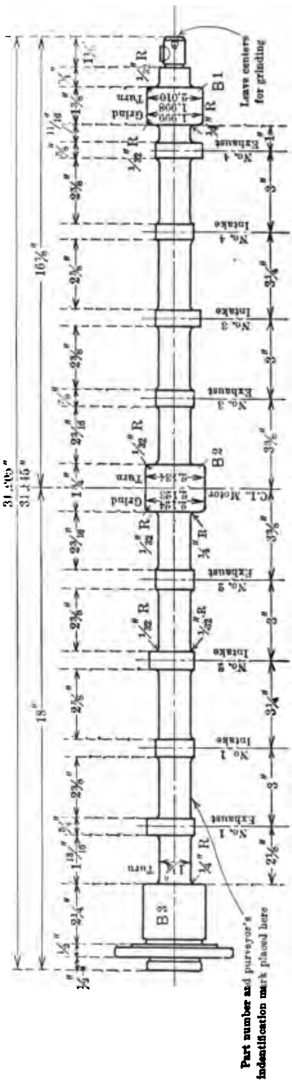
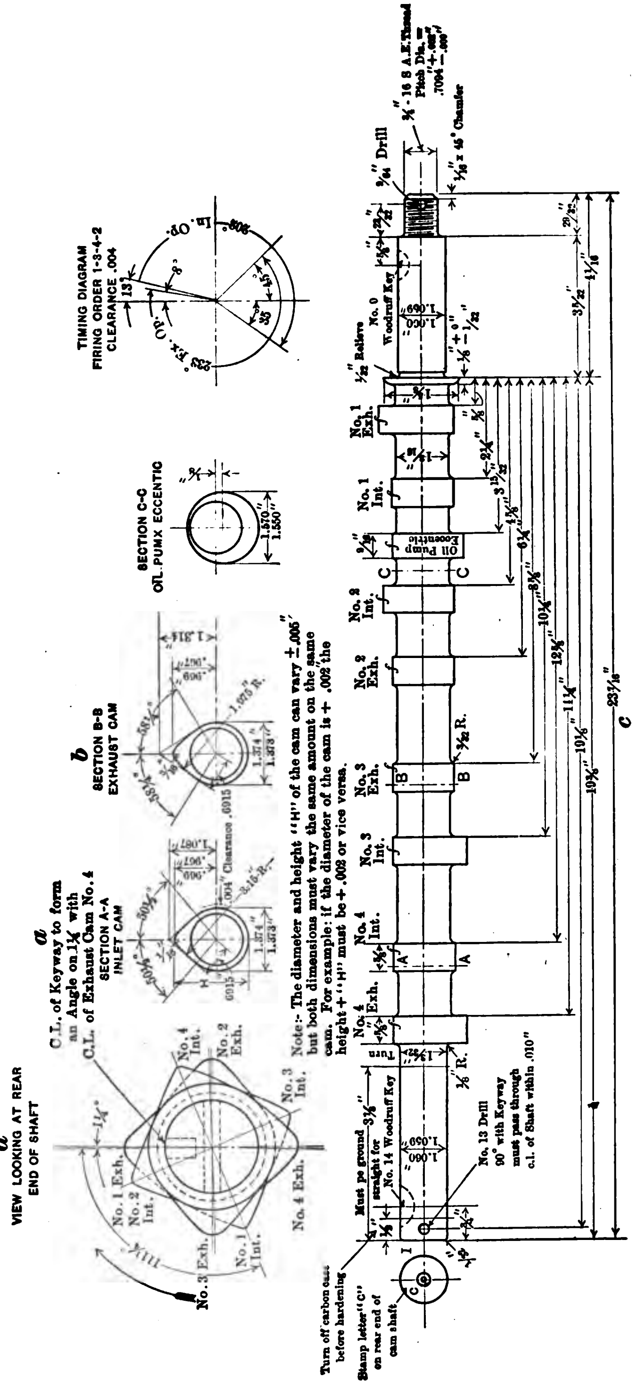


FIG. 59*g*.—Camshaft of Class B engine.

The exhaust valve opens (see timing diagram) 45° before lower dead center and closes 5° past upper dead center; therefore the exhaust will be open during the crankshaft motion of +45° + 180° + 5° = 230°, which corresponds to 115° on the cam outline.

Next we have to determine the position of the inlet and the exhaust cams with respect to each other on the camshaft. Beginning with the top dead center as 0°, the center of the inlet cam is 15° + ½ of time inlet is open; thus $15° + \frac{200}{2} = 115°$. The exhaust will open on the second revolution of the crankshaft, 45° before lower dead center or 180 − 45 = 135° from the top, and by adding one-half of the time it remains open we have 135 + 115 = 250°. It is necessary to add one complete revolution or 360°; therefore the center of the exhaust is 250 + 360 = 610°. These distances are on the crankshaft; for the cam shaft these figures are divided by two (as it rotates at one-half the speed) we have therefore ½ of 115° = 57°30′ for the center of inlet cam, and ½ of 610° = 305 for the center of exhaust cam. The



FIGS. 60, 60a, 60b, 60c, AND 60d.—Timing diagram, cams, and camshaft of the Briscoe motor (4-cylinder, $3\frac{3}{16}'' \times 5\frac{1}{8}''$).

difference is 305 − 57°30′ = 247°30′ on one side, and 360 − 247°30′ = 112° 30′ on the other side (see Fig. 59*d*).

Naturally, the center positions of all the inlet cams and all the exhaust cams are 90° apart in a four-cylinder engine.

Figure 60 shows the timing diagram of the Briscoe Motor. Figures 60*a* and 60*b* show working drawings of the convex flanked cams used on this motor.

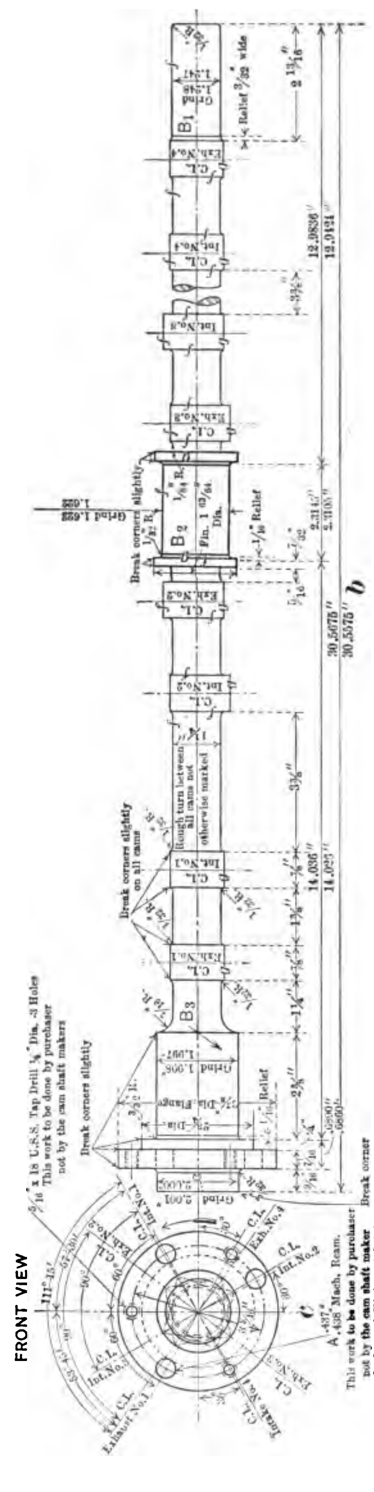
Figure 60*c* shows working drawing of the camshaft and Fig. 60*d* an end view of the shaft with cams in position. Figures 61 and 61*a* show inlet and exhaust cams with valve timing of the Hercules Motor. These are convex flanked cams used with the mushroom type of cam follower. It is noted that for designing the cam a clearance of .010 inch was assumed. Therefore, in addition to the valve action the angle will have to be increased on account of the clearance as shown on the diagram. The inlet valve here has a total angle of action 105° (210° on the crankshaft), while the exhaust has an angle of action 112°-30′ (225° on the crankshaft).

Figures 61*b* and 61*c* show the cam shaft details as required in working drawings. After the shaft is turned the cam surfaces are hardened and are then ground in a cam grinding machine. The necessity for having the cams hardened is obvious. Frequently the small gears driving the accessories, are also made integral with the shaft.

The diameter of the roller with the roller type of cam follower (used with the tangential cam) is limited by the diameter of the push rod; on the other hand a larger roller will work more satisfactorily. The roller diameter should not be less than about 1 inch, and the shaft of the roller not less than one-third, the diameter of the roller.

The width of the roller is made slightly less than that of the cam. The cam width, when used with roller, should be about .35 times the valve diameter. With the mushroom type of follower the cam width should be from .4 to .5 times the valve diameter. The force required to open the valves is quite considerable, especially when opening the exhaust, whose head must be lifted against the gas pressure at the end of the stroke, in addition to the spring pressure. Therefore the camshaft diameter must be such as to have the requisite stiffness, else it will spring too much in action.

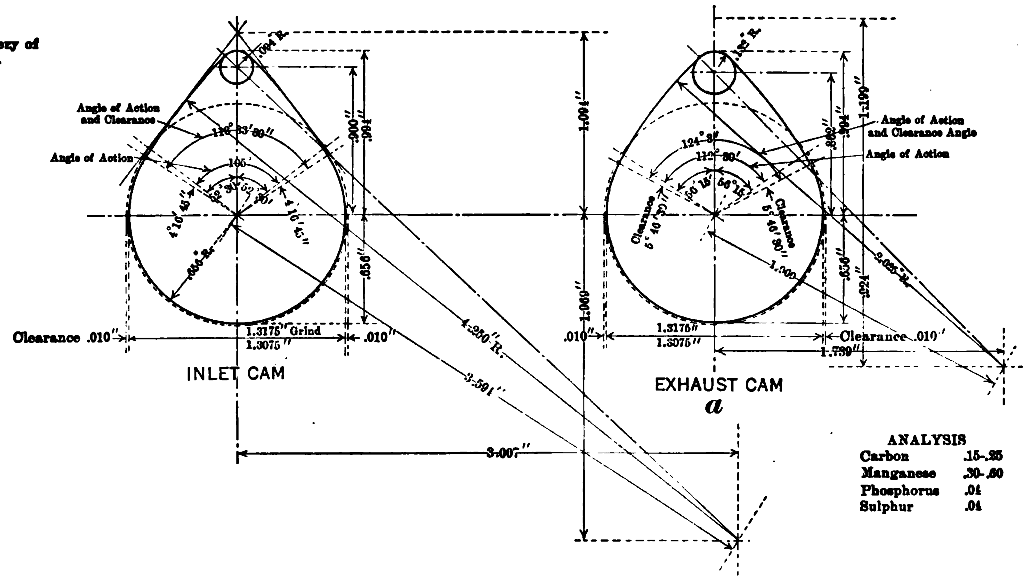
The diameter of the camshaft will also depend on the distance between the bearings.



FIGS. 61b AND c.

NOTE: All points on periphery of cams to test as follows.
Scleroscope 30-90
Brinnel 534-600

NOTE: Material 15 — .25 carbon steel, *D*, forging.
B. Hardness: 534 — 600.
S. Hardness: 80 — 90.



FIGS. 61, 61*a*, *b* AND *c*.—Valve timing diagram, cam and camshaft of Hercules motor Model M (4″ to 4½″ bore, 5½″ stroke).

In present designs as a rule three bearings are provided for four-cylinder engines, likewise for six-cylinder motors, especially the small powered sixes. With larger engines, four or five bearings are employed while with small engines sometimes only two. (See Figure 60).

Present day practice favors the insertion of the camshaft into the crankcase from the front end; the case being provided with bosses which have bored and reamed holes (often termed tunnels) see Y, Figs. 34, 35, 36, 36*a*, in which the camshaft bearings are

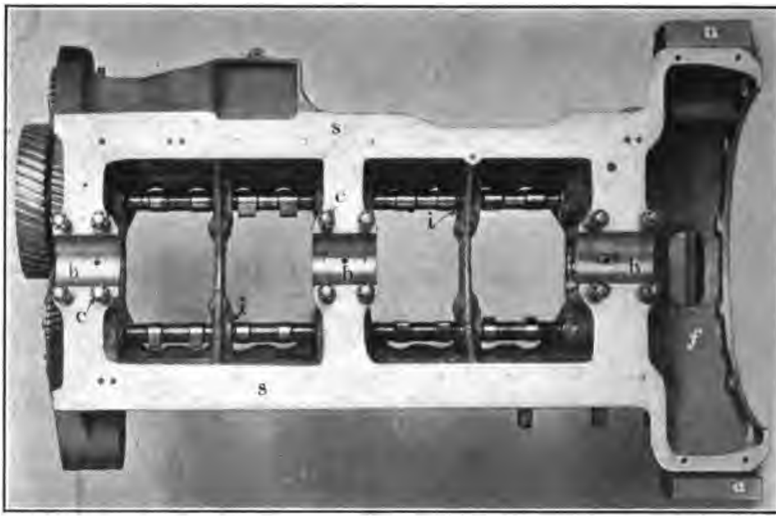


Fig. 62.—Inverted plan view of upper half of crankcase of the White 16-valve, 4-cylinder engine. (Its manufacture has been discontinued).

securely held by a press fit, or they are held in another support, see. Fig. 36*a*. Instead of making a shaft separately and attaching cams thereto as was done formerly, the camshaft is usually drop forged of one piece integral with the cams, as seen from Figs. 59, 60, and 61. Figure 62 shows an inverted view of the crankcase of the White 16-valve, four-cylinder engine with the camshafts in position. This 16 valve engine is not now manufactured but it illustrates a good example of this type.

The journals *B*1, *B*2, and *B*3, of the shaft (Fig. 59*g*) are slightly larger than the cam height so that the whole shaft can be inserted or withdrawn through the bearings of the crankcase. The bearings are ordinarily provided with bronze bushings, as at *E*, Fig. 36*a*.

Figure 35*a* shows at *F* how the bushings are made in the Haynes Light Six Motor. In this camshaft the journals are not enlarged therefore the bushings or bearings are made to fit the tunnel of the crankcase. To eliminate unnecessary weight note how they are constructed. At *G* the oil accumulates and flows through the hole into the large oil groove or container *H*.

The thickness of the metal is for the most part only 1/8 inch. The bushing is held in place by stud *I* provided with lock nut and washer. In the center bearing of the camshaft, unless the journal is enlarged, the bushing has to be made in halves.

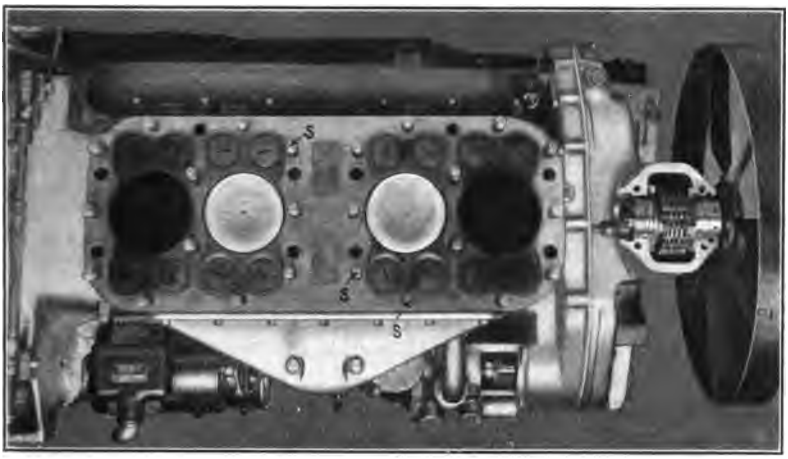


Fig. 62*a*.—Sixteen-valve, 4-cyiinder White engine with cylinder head removed. (Its manufacture has been discontinued).

Occasionally the front bearing is provided with logs or with a flange, whereby it is secured to the crankcase, as in *D*, Fig. 36*a*. The journals of the camshaft are frequently made of various diameters to facilitate its insertion in the crankcase, for instance, *B*1, Fig. 59*g*, which is at the end 2 inches (this could be made smaller and frequently is, as it does not have to clear the cams); *B*2 is 2 1/8 inches and the front bearing *B*3, 2 1/4 inches approximately. The front bearing to which the gears are attached for driving the camshaft as well as for the drive of the various accessories, is usually the largest. In many cases the difference in diameter of the bearings of a shaft are not as large as those given above. The bearing length is usually about the same or slightly larger than the bearing diameter, with the enlarged bearing type

of camshaft shown. When the journals are of the same diameter as the shaft the bearing length is made about three times the shaft diameter, if the distance between the bearings is more than 15 inches. If the distance is less the bearing length can be reduced. Frequently, the bearing to which the driving gears are attached, are made slightly longer while the others are made slightly shorter. Camshaft bearings are made of bronze or cast iron; sometimes of special babbitt alloys, or they are bronze backed babbitt lined. Bronze, being a good conductor, will assist in keeping the bearings cool.

In the Class B engine the bearing lengths are 1⅜, 1¾, and 2¼ inches from rear to front respectively. The Buda, Mack and Continental Engines of which we spoke in former chapters have camshafts and bearings of the following dimensions:

Buda Engine (4¼ × 5½ bore and stroke, valves 1⅞, and valve lift ⅜); having convex flanked cam and mushroom cam follower, camshaft diameter, 1 inch; cam bearings diameter 1½, 1¾, and 2 inches. Length of bearings, 1½, 1¾, and 2¼ inches.

Mack Engine (4 inches × 5 inches, valve diameter 1¾, lift 11/32 inch); tangential camshaft, diameter 1¼ inches, camshaft rear bearing 1½ inches, center bearing 2.17 inches and front bearing 2.185. Length of rear and middle bearings 2 inches, length of front bearing 2⅝ inches.

Continental Engine (3½ inches × 5¼ inches, valves 1½ inches, lift 5/16 inch approximately); mushroom type cam follower; camshaft diameter, 1¼ inches, cam bearing diameters about 2⅛ inches, width of bearings rear and front, 1¾ inches, center 1⅛ inches. It might be mentioned that the distance between the center of the bearings in the last three engines named varied from 13 to 15 inches. If this distance is longer the camshaft diameter is appropriately increased. (Camshaft diameter varies in engines from 3½-inch bore to 5-inch bore from ⅞ inch to 1⅜ inches ordinarily.)

The Briscoe being a small engine, and having bore and stroke of 3 3/16″ × 5⅛″, and a valve diameter of 1 7/16″, there are only two camshaft bearings (see Fig. 60*c*), but the shaft length from center to center of bearing is only about 17¼″ and the shaft diameter 1 3/16″ is sufficient for such a small engine.

In Fig. 62 (White 16-valve engine) there are five bearings in each camshaft, *i.e.*, in addition to the two end bearings and the center bearing, there are two intermediate bearings. Fig. 62*a*

is a view of this engine with the cylinder head removed, showing the sixteen valves of this four-cylinder motor. In addition to making possible a better volumetric efficiency without abnormally large valve pockets, by using four valves in each motor, the valves are smaller in diameter and can therefore be kept at a lower temperature. All the exhaust valves are located on the top of the illustration and the inlet valves at the bottom.

CHAPTER VIII

CAMSHAFT AND ACCESSORY DRIVE

The camshaft is driven by the crankshaft, and since in a four cylinder engine each valve is operated only once during two revolutions of the latter, the camshaft must rotate at one-half crankshaft speed, and its gear therefore, has one-half the number cf teeth of the driving gear attached to the crankshaft.

The magneto must furnish a spark once in each cylinder in two revolutions of the crankshaft, and practically every magneto made furnishes two sparks per revolution of its shaft. For a two cylinder engine, such a magneto has to be driven at one-half crankshaft speed, for a four-cylinder motor at crankshaft speed, and for a six-cylinder, at one and one-half times the speed of the crankshaft. Hence in a four cylinder engine the magneto gear has the same number of teeth as that of the crankshaft, while in a six-cylinder it has only two-thirds that number.

The fan is usually driven at twice the speed and the water pump at one and one half times the speed, more or less, depending on the pump capacity (see Water Pumps).

The camshaft as well as the accessories are generally driven by the crankshaft pinion as illustrated in Fig. 63 to 70. The driving gear attached to the crankshaft is ordinarily made of steel, while the gear of the camshaft, as well as those of the accessories, are made of cast-iron, sometimes of bronze or of fibre compositions.

To reduce the noise of the gears, they are mostly cut with helical teeth having a tooth angle of about 30 degrees. (Sometimes 27°, and in some engines only 20°.)

The face width of the crankshaft and camshaft gears are made alike, but the width of the generator gear and those of other accessories are frequently narrower, since the driving power for these is not so great. The width of the crankshaft and camshaft gears are approximately ¾ inch for a 3 inch bore engine, increasing by ⅛ inch for each one inch of bore. To express it in a formula we have f, face width $= .75 + (b - 3)\ .125$ inch, for touring cars, and for truck motors, $f = .75 + (b - 3).175$, may be used. ($b - 3$ is the increase above 3 inches).

When a 20 degree tooth angle is used the face width is made somewhat larger for small engines than that derived from the formula, in order to have at least one tooth fully in mesh all the time.

Figures 59*c* and 59*d* show the holes provided in the flange of the camshaft to which the gear wheel (in mesh with the crankshaft pinion) is attached. Frequently the shaft is provided with a woodruff key (Fig. 60*c*) instead of a flange for its attachment to the gear wheel.

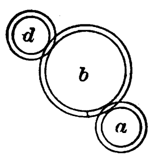


FIG. 63.

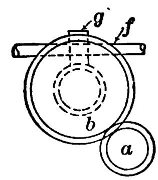


FIG. 64.

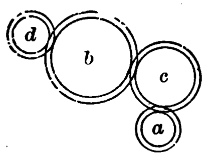


FIG. 65.

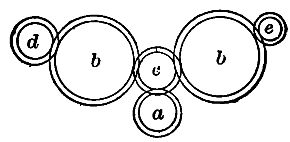


FIG. 66.

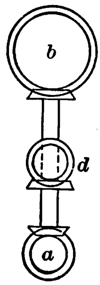


FIG. 67.

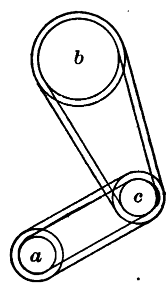


FIG. 68.

FIGS 63, 64, 65, 66, 67 and 68.—Types of camshaft and accessory drives.

Figure 63 shows an arrangement for driving an L-head motor where the motor valves are located at one side of the engine. In this type both the magneto and the pump are generally located at the valve side; sometimes, however, at the end of a cross shaft in front, as shown in Fig. 64. This is the system used in the Mack Engine referred to. Here the magneto and the pump are driven at the ends of the cross shaft *f* which is situated in front. The camshaft is driven directly through spur gears and

on the camshaft is mounted a helical gear which meshes with another helical gear *g* on the cross shaft above it.

In the Figs., *a* represents the crankshaft pinion; *b* represents the camshaft gear to which the magneto and the pump are connected.

Frequently an idler *c* is used (Fig. 65), between the pinion and the cam gear. When no idler is used it requires larger gears as the distance between crankshaft and camshaft is frequently quite considerable. It should be remarked that an idle wheel between the camshaft and the crankshaft reverses the direction of the former from that which it would be if there were no idler.

Figure 66 shows an arrangement for a T-head motor. In this

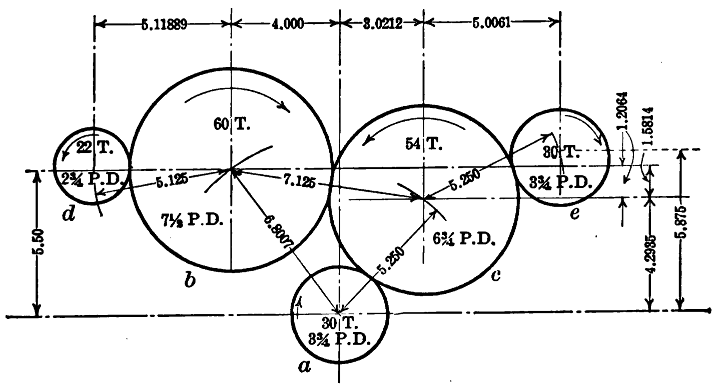


FIG. 69.—Camshaft and accessory drive of Class B, military truck engine.

case there are two camshafts, one at each side of the cylinders, and *d* and *e* are the magneto and the pump gears respectively. Since the pump is driven at a higher speed, the diameter of the gear wheel is proportionately smaller than the crankshaft pinion. Overhead camshafts are driven by gears or by means of chains, as shown in Figs. 67 and 68; *d* in Fig. 67 is the shaft, driving magneto and pump.

Figure 69 shows the type of camshaft and accessory drives used by the Class B Military Engine. In this case the drive is from the crankshaft gear "*a*" through an idler "*c*" to the camshaft gear "*b*"; the latter drives the generator drive gear "*d*," while the idler drives the water pump gear "*e*." This diagram also shows the gear centers with relation to each other, in inches; it also gives the number of teeth in each wheel, and the pitch

diameter. For instance, the crankshaft gear has 30 teeth and a pitch diameter of 3¾ inches; the camshaft gear "*b*" has double the pitch diameter and double the number of teeth.

The timing gears are 1¼ inches wide, with teeth cut on an angle of 27¼°.

Figure 70 shows a typical silent chain drive manufactured by the Morse Chain Company, with adjustable sprocket on the accessory shaft *d*. This concern states that in the ordinary car of to-day it is undesirable to use more than 24 teeth in the crank-

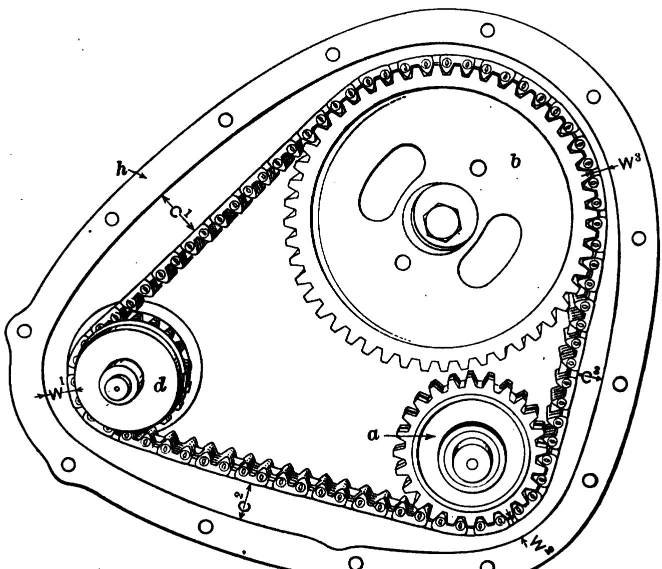


FIG. 70.—Silent chain drive.

shaft sprocket, which is 1 foot chain speed per r.p.m. of the motor. It is also undesirable to use fewer than 18 teeth in the crankshaft sprocket. A driven sprocket, as used for the pump, for instance, is permissible as small as 14 teeth, but nothing less. Figure 70 and the following table show the clearance recommended by the Company named, between the crankcase casting *h* and the chain, and between the sprocket wheels and the case. This silent chain is very largely used abroad for L-head engines where the pump and the magneto are located at opposite sides from the valves.

RECOMMENDED CASE CLEARANCE

P = Pitch of chain or $\frac{1}{2}''$

Wheel (from top of tooth)

$W^1 = P + \frac{1}{8}'' +$ Adjustment $= 1\frac{5}{16}''$

$W^2 = P + \frac{1}{8}'' = \frac{5}{8}''$

$W^3 = P + \frac{1}{8}'' = \frac{5}{8}''$

Chain (from top of chain)

$C^1 = P \times 2\frac{1}{2}'' = 114''$

$C^2 = P \times 3'' = 1\frac{1}{2}''$

$C^3 = P \times 2 = 1'$

(C^1, C^2 and C^3 entirely depends on the length of strand.)

CHAPTER IX

PISTONS

The crankshaft of a motor receives its impulses from the piston, which in turn receives the force from the expanding gas. Most of the pistons in use are made of a good quality, close grained, cast-iron, though the last few years a number of manufacturers have adopted aluminum alloy pistons. In high-speed engines, it is very important that the pistons and the connecting rods should be made as light as possible in order to eliminate vibration due to the inertia effect. For this reason, it is advisable to machine the piston on the inside, wherever possible, as well as on the outside; this reduces the piston weight, and insures a more uniform weight of all the pistons of an engine. If some are heavier than others, it is conducive to vibration on account of their not being in balance, and as the castings cannot be made absolutely accurate or uniform, it is best to machine them all over, especially high-speed pistons.

To make the cylinder gas tight requires expanding piston rings on the piston, since the latter cannot be made to have a pressure tight fit with the cylinder. It is necessary to allow a certain amount of clearance between the piston and the cylinder and to provide piston rings, which expand and which are placed on the piston in grooves turned for the purpose. There are two general methods used for the manufacture of rings; one is to make the ring the same size as the bore of the cylinder and peen it, to give it the desired spring, that is to say, hammer the inside of the ring so that it springs open; the other method is to make the ring somewhat larger in diameter than the bore, then cut a small piece out and finish it while in a compressed state to the size of the bore.

Some manufacturers prefer not to grind the outside of the ring but to leave the last fine cut from the lathe tool on it; after the piston has run for some time, it will wear 'smooth' inside the cylinder. Most manufacturers however, grind their rings on the outside. To place the ring in position it is sprung open

until it slides over the piston and is pushed down until it reaches the ring groove, when it will spring into place.

As a rule three piston rings are used, sometimes only two and occasionally four. With four rings (at times also when three are used), frequently the lowest ring is placed near the lower end of the piston to act as an oil scraper ring. The material used for the rings should not be harder than that of the cylinders else it will cause extra wear to the latter; for this reason cast-iron is as a rule employed.

The width of the rings depends somewhat on their number, yet there is a tendency among designers to make them a certain width irrespective of their number and varying them only with variation of bore. The width w may be made $\frac{1}{8}$ inch for pistons up to $2\frac{1}{2}$ inch bore, widening them $\frac{1}{32}$ inch for each $\frac{1}{2}$ inch increase of bore. We may write therefor $w = .125 + (b - 2.5)$ $.062$. While the thickness t of the ring is approximately $t = \frac{b}{30}$, at the thickest portion, gradually decreasing toward the split portion in eccentric rings to $\frac{2}{3}t$. Means are sometimes provided for preventing the rings from rotating in their grooves, but on account of the difficulties, of pinning them in the thin castings, they are as a rule dispensed with. Many designers use the following dimensions for the width of rings, which correspond closely to the formula just given.

For cylinder bores of from 2″ to $2\frac{1}{2}$″, width of ring $\frac{1}{8}$″.
For cylinder bores of from $2\frac{5}{8}$ to 4″, width of ring $\frac{3}{16}$″.
For cylinder bores of from $4\frac{1}{8}$″ to 5″, width of ring $\frac{1}{4}$″.
For cylinder bores of from $5\frac{1}{8}$″ to 6″, width of ring $\frac{5}{16}$″.

As regards piston clearance, with cast-iron pistons, it is usual to allow (to make piston smaller) about .003 (.004 for trucks) per inch of cylinder bore for the head of the piston, .002 per inch of bore immediately below the head and .001 per inch at the bottom end of the piston. Sometimes the piston is tapered from the head down toward the bottom end, at other times, it is made smaller in laps or lands until the piston rings have been passed, and then the skirt is tapered. The bottom of the piston skirt is frequently provided with a few grooves around its periphery in which oil can collect. While the piston ring sometimes used at the bottom scrapes the oil back when an excessive amount tends to travel up into the combustion chamber. During the suction stroke (especially with a partly closed throttle) there is

a tendency for oil to be drawn up into the combustion chamber through the clearance space between piston and cylinder. A ring at the bottom will also reduce piston slap.

There is a side thrust of the piston against the cylinder wall, therefore the piston must be of sufficient length that the friction between it and the cylinder side walls be not excessive. From 25 to 30 lb. during the power stroke is taken as the mean side thrust per square inch of projected area. (Projected area = length × diameter.) The length of piston is, as a rule, from 1.2 to 1.3 times the bore, while aluminum alloy pistons are made from 1.3 to 1.33 times the bore. It should be borne in mind that if 25 or 30 lb. is taken as the mean pressure per square inch of projected area, when figuring the length of piston the space occupied by the rings must be deducted, also the length of the under-cut portion near the piston-pin axis, when this under-cut portion is carried all around the piston. It is customary to under-cut the piston, that is to say, make it slightly smaller in diameter where the piston pin is, and for a certain length on each side of the pin axis. This is done to prevent any deformation of the piston when as a result of the pressure exerted on it the pin is slightly flexed. The length of this under-cut portion u (Fig. 72) is frequently made .3 times the bore and is made about $\frac{1}{32}''$ smaller in diameter than the piston. In the latest designs it is not carried all around the piston but is made as shown in Figs. 73 and 74. In some designs it is almost eliminated (see Fig. 75). The smaller the friction between piston and cylinder, the longer is the life of the cylinder, for this reason, in truck engines the pistons are frequently made longer. Occasionally they are 1.5 times the bore.

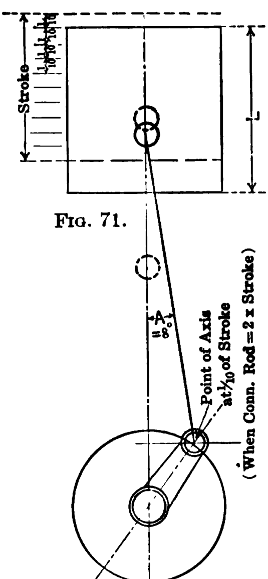


FIG. 71.

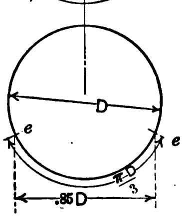


FIG. 71a.

The pressure with which the piston bears against the side of the cylinder wall at any particular instant can be determined

by the equation $B = P \times$ tangent A, where P is the pressure in pounds on the piston head for any particular point of the stroke, B is the total pressure between piston and cylinder, and A the angle formed by the connecting rod with respect to the center line of the cylinder (see Fig. 71).

The pressure against the cylinder reaches its maximum when after the ignition of the gases, the piston has traveled about one-tenth of its stroke. At this point we may take the maximum pressure in the cylinder as 300 lb. per square inch. If the connecting rod length is, let us assume, twice the length of the stroke, the angle A can easily be found graphically by making a drawing, and we find that this angle is approximately 8°. The tangent of $8° = .14054$. This value can be found from any book giving trigonometrical tables. We then have for maximum

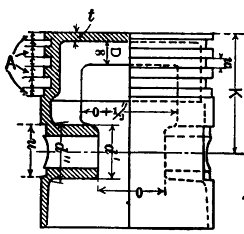


FIG. 72.

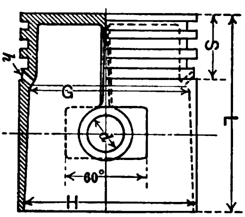
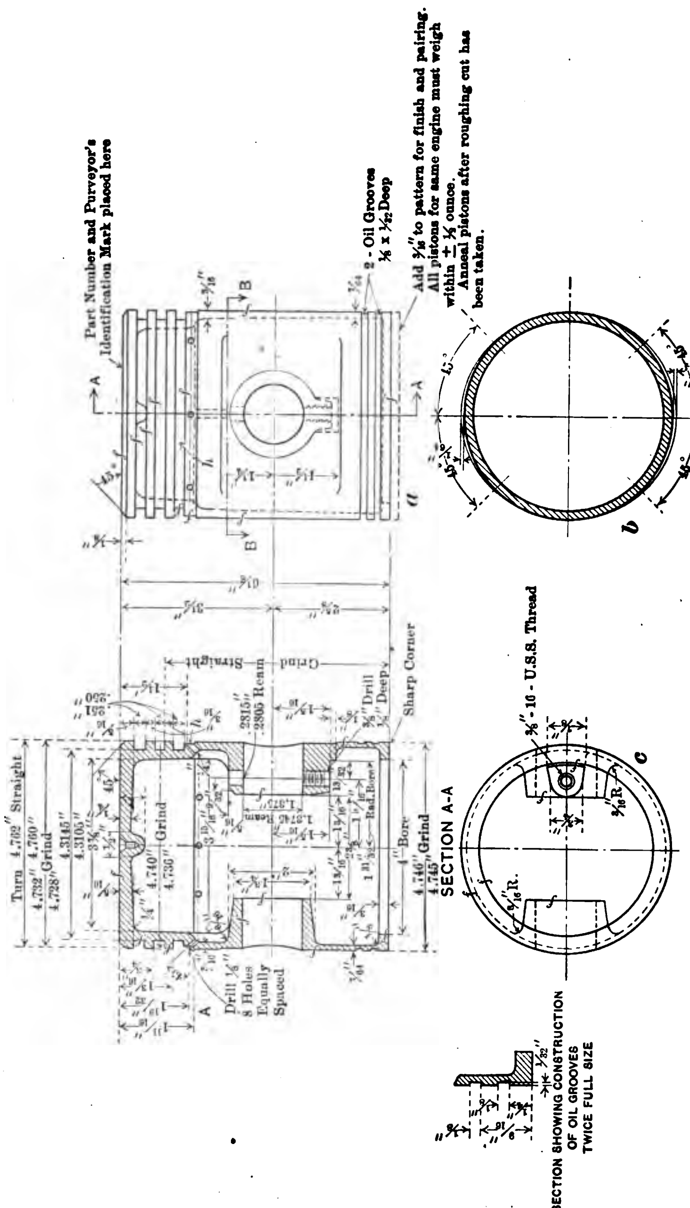


FIG. 72a.

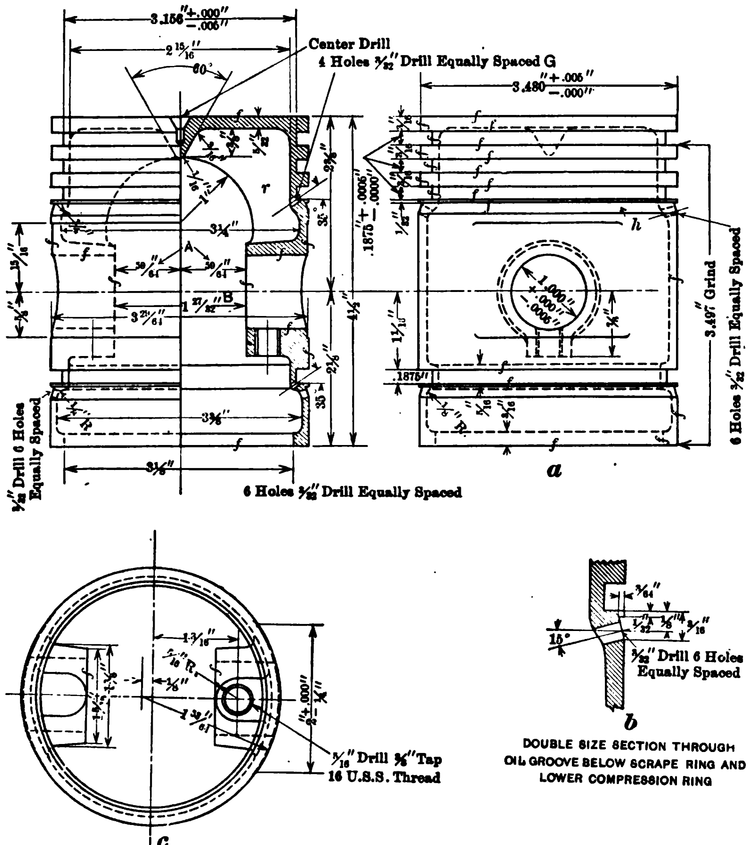
friction pressure B between piston and cylinder, the following: B = area $\times$ pressure per square inch $\times$ tangent A. Thus $B = \frac{\pi}{4} \times D^2 \times 300 \times .14054 = 33D^2$ approximately, where D is the diameter of the cylinder, *i.e.*, the bore. Evidently, the longer the connecting rod, the smaller would be the angle, and therefore the value so found would be smaller. We assume that the piston pin is in the center of the bearing length of the piston (the bearing length of the piston is the total length of the piston minus the space occupied by the rings and the under-cut portion, if there is any all around the piston); the actual width of the bearing surface is in reality not the projected area but one-third the circumference (see Fig. 71*a*). By projecting this on the circle (*ee* represents one-third of the cylinder circumference), the distance measured straight across from *e* to *e* will be found to be equal to $.85D$. The length of the bearing surface of the piston, let us assume, is $.8L$; L representing the



FIGS. 73, 73*a*, 73*b*, AND 73*c*.—Piston of Class B military truck engine.

total piston length, then the maximum pressure p per square inch on the bearing surface equals $p = \frac{33D^2}{.85D \times .8L} = \frac{43.5D}{L}$.

This gives us the maximum pressure To obtain the mean pressure a number of angles A have to be taken along the entire piston travel, when the mean pressure can easily be determined.

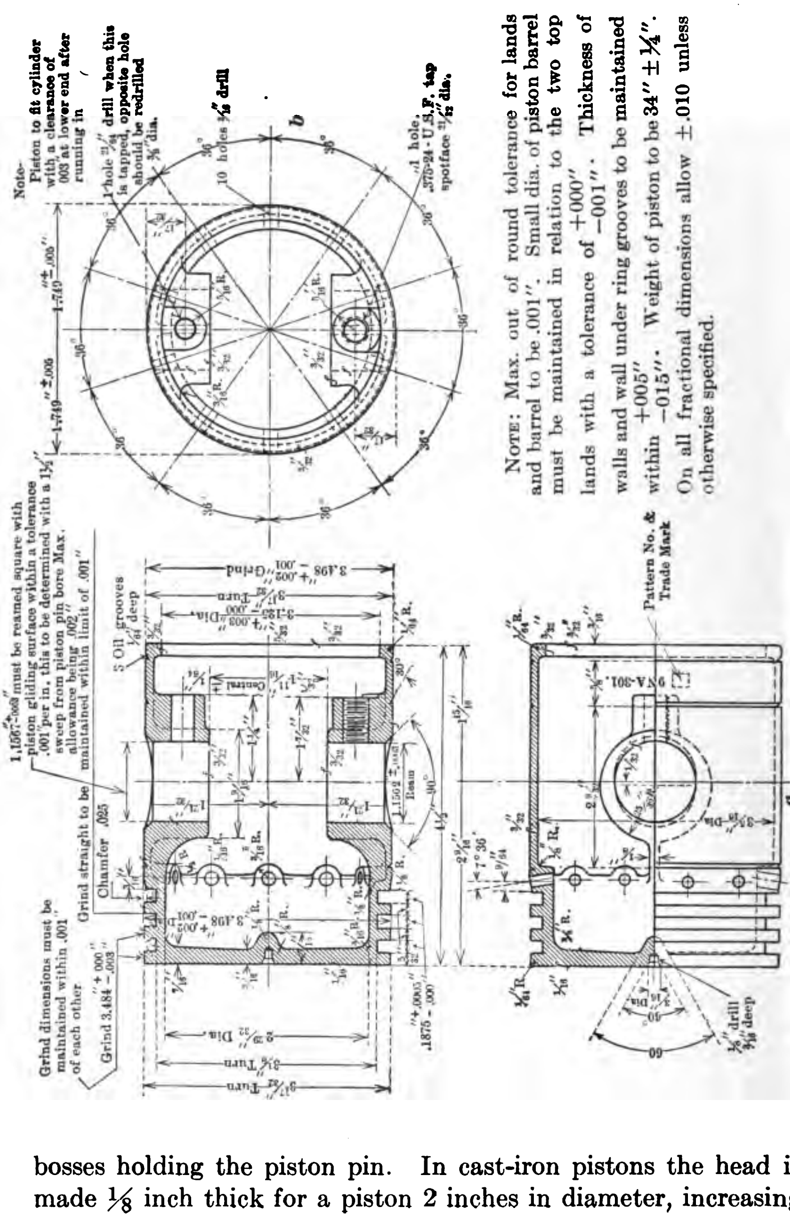


FIGS. 74, 74*a*, 74*b*, and 74*c*.—Piston of Haynes light-six engine.

For instance, if we find the pressure p for 10 equi-spaced positions of the piston, by adding all the pressures so found and dividing the same by 10, we obtain the mean pressure.

The thickness of the piston head depends on the diameter b of the piston and on the material used, and also on the ribs provided

on the under side of the piston head. As a rule, the ribs, when such are provided (Fig. 74), extend from the piston head to the



FIGS. 75, 75*a* AND *b*.—Cast iron piston of Continental Model 9N six cylinder engine (3½" × 5¼").

bosses holding the piston pin. In cast-iron pistons the head is made ⅛ inch thick for a piston 2 inches in diameter, increasing

by $\frac{1}{32}$ inch for each one inch increase in the bore, that is to say, the thickness t is as follows:—

$$t = .125 + (b - 2)\ .031.$$

In touring cars when the piston head is well ribbed it may be made $t = \frac{3}{32} + (b - 2).025$ inch, and frequently even less. The piston head is frequently made tapered, being thicker near the outer circumference. An extra $\frac{1}{16}''$ is added to the first formula given above. In other words, the head thickness t where it joins the piston barrel $= .125 + (b - 2)\ .031 + \frac{1}{16}$. When no ribs are provided as in Fig. 73, which is the piston used on the Class B. Military Truck Engine, the thickness of the head $= \frac{5}{32} + (b-2)\ .031$ near the center. (For higher speed engines, the piston should be made somewhat lighter.) This engine has a bore of 4.75, the piston head according to our formula would therefore be $\frac{1}{4}$ inch. It tapers toward the circumference to $\frac{5}{16}$ inch.

The depth of the grooves receiving the piston rings should be at least $\frac{10}{1000}$ greater than the maximum thickness of the piston rings. (See table for groove depth, page 111.)

The piston wall behind the rings should be from $\frac{1}{64}$ to $\frac{1}{32}$ inch thicker than the top of the piston skirt, see Figures 73 to 75. The thickness of the top of the piston skirt is usually $= (b - 2)\ .02 + .1$ inch, tapering at its lower end to from one-half to three-quarters the thickness of its upper end. (The piston skirt is the piston barrel below the rings.) In truck engines this dimension may be slightly more, and in automobile engines slightly less. In Fig. 73*a* it is shown dotted to taper from $\frac{3}{16}$ inch to $\frac{7}{64}$ inch. In Fig. 74 which is the piston used in the Haynes Light Six Engine, the piston skirt tapers below the ring from $\frac{1}{8}$ inch in thickness to $\frac{1}{16}$ inch. By looking at this figure it will be noted that the inside diameter of the piston below the rings is $3\frac{1}{4}''$, while near the bottom it is $3\frac{3}{8}''$. Frequently at the lower end of the piston skirt a small internal flange is provided to strengthen the piston where the wall is very thin.

Aluminum Pistons.—In aluminum pistons the clearance required is usually $.005b$ (piston diameter is made $b - .005b$) at the top land *i.e.*, at the piston head, and $.002b$ at the top of the piston skirt, and the clearances of the several lands are between these two values, while the lower end of the skirt is $.0015b$. In order to facilitate the transfer of heat in aluminum pistons from the head down to the barrel or skirt, occasionally internal

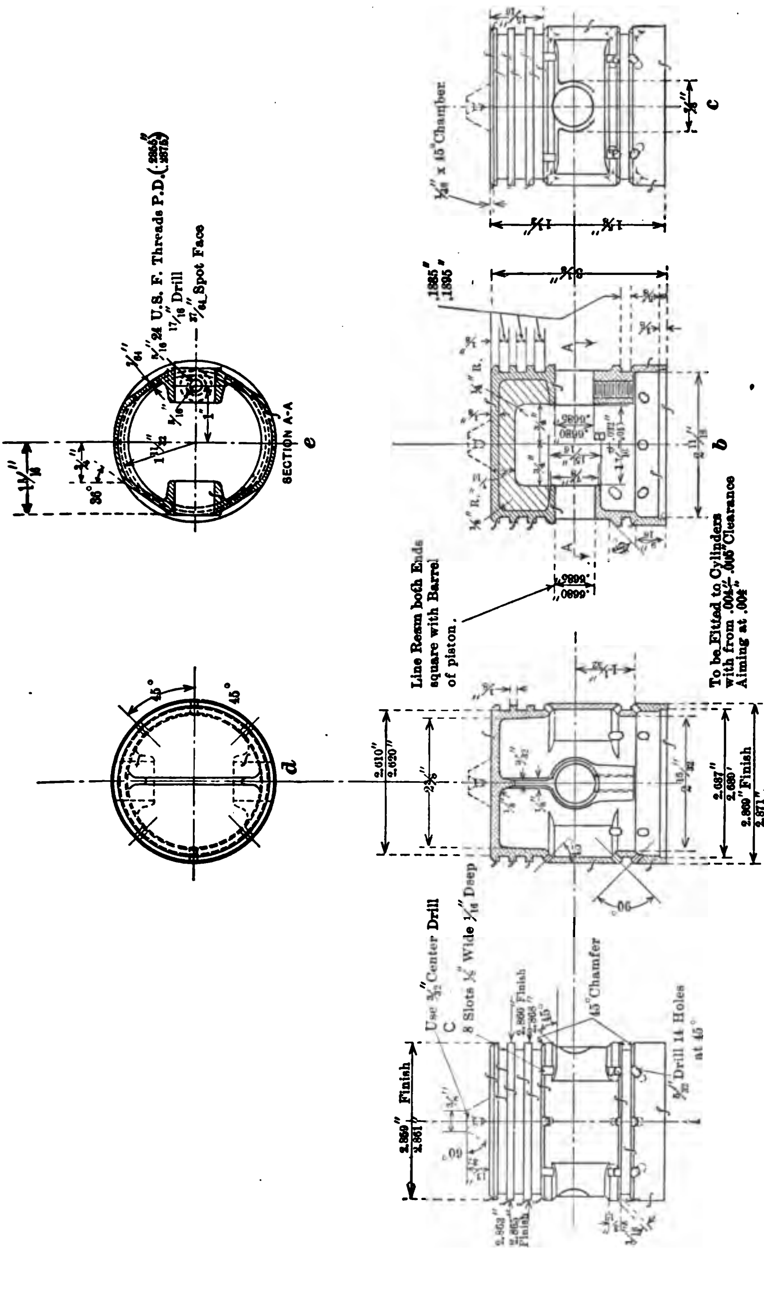
ribs at right angles to the piston bosses are provided. Alloy pistons, when well ribbed, will not expand nearly as much, as they can be kept at a lower temperature, and therefore smaller clearances are permissible.

When determining the thickness t of the piston head, with alloy pistons, it must be borne in mind that aluminum has a very much lower melting point than cast iron (melting point of cast iron 2100°F., of aluminum 1200°F.), and that metals lose some of their strength when frequently heated to a comparatively high temperature with respect to their melting point. For this reason aluminum pistons will have a reduced strength when operating at the high temperature usually prevailing in internal combustion engines.

Therefore, it is advisable to make the piston head slightly thicker than is required for cast-iron pistons. (Some designers prefer to provide extra ribs instead.) A convenient formula may be expressed as follows: $t = 3/16 + (b - 2)\ .031$ inches. When the head is ribbed and especially in small bore pistons, t may be made the same as for cast iron (see Figs. 76 to 76*e*), which is the die cast aluminum piston used on the eight-cylinder ($2\frac{7}{8} \times 4\frac{3}{4}$) Oldsmobile (Northway Engine). These are the complete working drawings of the piston of this good engine and its careful examination is recommended to students.

On account of the increased clearance with aluminum pistons, some trouble is experienced with "oil pumping," that is to say, the suction of oil into the combustion chamber, when the engine is first started or when it is running throttled down, and thus having greater suction.

Another cause of trouble is piston slap, that is to say, the piston, before reaching the temperature at which it will fully expand, will hit the side of the cylinder, and this is called the piston slap. To prevent oil pumping into the cylinder, with aluminum as well as cast-iron pistons, a shallow groove is turned on the outside of the piston (see h in Figure 72), beneath the lowest ring; it is turned either square as shown in Figures 72 and 73, or simply tapered off from the outer circumference toward the piston ring groove as seen from Figures 74 to 76, and a number of holes are drilled through the piston wall, sloping downward, as shown. The number of the holes vary in the various designs from 6 to 10, and their diameter ranges from 3/32 inch to 3/16 inch. A good plan is to have some of these small



FIGS. 76, 76*a*, *b*, *c*, *d* AND *e*.—Die-cast aluminum alloy piston used on the eight-cylinder Oldsmobile (Northway engine, $2\frac{7}{8}'' \times 4\frac{3}{4}''$).

holes reach the top of the piston-pin bearing (when the pin revolves in the piston).

With alloy pistons one or more deep ribs across the inside of the head will also assist to keep the piston cool. On account of the increased heat conductivity of aluminum alloys the piston head will be cooler and the compression pressure can be increased somewhat for engines with small bore. There will also be less carbon deposit on the piston head, and no carbon deposit on the underside of the piston head. Furthermore these alloys being softer than iron or steel, the cylinder itself will not show so much wear, so that in case of piston seizure, the piston and not the cylinder is scored. Aluminum pistons are frequently cast in metal dies when very little machining has to be done. The coefficient of expansion for aluminum is .00023, and for cast iron, .0000119 per degree Fahrenheit, which accounts for the increased clearance required, but on account of the greater conductivity, the aluminum piston head will not be of so high a temperature, and therefore the difficulty due to expansion is not as great as it would appear. The great advantage with aluminum pistons is of course the reduced weight for high-speed engines. Many makers, however, use alloy pistons which are almost as heavy as cast iron, but on account of the increased thickness of metal and the greater conductivity of the metal, it reduces the piston temperature and therefore improves the lubrication between piston and cylinder. The reduction in weight which can be secured with alloy pistons is an important factor by insuring smoother running, and less vibration. The specific gravity of aluminum alloy is in the neighborhood of 2.7 while that of cast iron is 7.25. In other words, one (1) cubic inch of cast iron is almost three times as heavy as that of aluminum.

Another tendency in piston designs is to relieve the skirt of the piston somewhat on the non-bearing side. This practice has been followed for years with general bearings where there is no appreciable load on the sides of the bearing in line with the bearing axis, and it is found that there is less friction when the sides are relieved. The film of oil which is sheared as the shaft revolves does not extend all around. Since the oil film at the side does not reduce the pressure on the portion of the bearing carrying the load it simply produces a loss due to the shearing of the oil. For the same reason, by relieving the piston skirt at the non-bearing side, there is less friction.

When machining aluminum pistons a mixture of kerosene with 10 per cent. lubricating oil is found to act well as a lubricant.

Figures 73, 73*a*, 73*b* and 73*c* show the complete working drawings of the Class B engine piston and attention is drawn to the absence of a rib underneath the piston head. For this reason the greater part of the inside of the piston can be turned if desired. A very small rib only is provided about the wrist-pin bosses. Figures 74 and 74*a* show four rings, one below the wrist pin. In this piston there are a larger number of oil holes than are usually provided; for instance, there are four in the groove of the third ring; underneath this ring there are six; and another six holes are slightly below the ring as seen as seen in Fig. 74*b*.

The Society of Automotive Engineers recommends the following data for piston ring width, width of grooves, etc., and designers are advised to make use of these "Standards." The following is the formula by which the piston ring grooves were calculated:

$$G = \sqrt{\frac{.01D^2}{8}} + .005 \text{ where } D \text{ is the piston diameter.}$$

TABLE III

Dia. (D)	Depth G	Diameter and limits Bottom ring groove		Groove width	Ring width
2	.1000	1.7960	1.790–1.796[1]	.1250[2]–.1260	.1237–.1247
2¼	.1064	2.0347	2.029–2.035	.1250–.1260	.1237–.1247
2½	.1132	2.2686	2.263–2.269	.1250–.1260	.1237–.1247
2¾	.1202	2.5041	2.499–2.505	.1875–.1885	.1862–.1872
3	.1275	2.7390	2.733–2.739	.1875–.1885	.1862–.1872
3¼	.1349	2.9737	2.968–2.974	.1875–.1885	.1862–.1872
3½	.1425	3.2080	3.202–3.208	.1875–.1885	.1862–.1872
3¾	.1503	3.4419	3.436–3.442	.1875–.1885	.1862–.1872
4	.1581	3.6758	3.670–3.676	.1875–.1885	.1862–.1872
4¼	.1661	3.9093	3.904–3.910	.2500–.2510	.2487–.2497
4½	.1741	4.1428	4.137–4.143	.2500–.2510	.2487–.2497
4¾	.1822	4.3761	4.371–4.377	.2500–.2510	.2487–.2497
5	.1904	4.6092	4.604–4.610	.2500–.2510	.2487–.2497
5¼	.1986	4.8423	4.837–4.843	.3125–.3140	.3110–.3120
5½	.2069	5.0752	5.070–5.076	.3125–.3140	.3110–.3120
5¾	.2152	5.3081	5.303–5.309	.3125–.3140	.3110–.3120
6	.2236	5.5408	5.535–5.541	.3125–.3150	.3110–.3120

[1] Preferred maximum dimension.

[2] Preferred minimum dimension.

THE WRIST PIN OR PISTON PIN

Hitherto practice has been either to fasten the pin into the piston bosses so that the pin rocked in the upper bearing of the connecting rod or else to lock the pin and rod together and rock the pin in the piston. The latter is probably a somewhat cheaper construction and permits a slightly longer bearing length. In practice the wrist-pin bearing pressure varies between 2500 and 3000 pounds per square inch of projected area, although in some cases a higher pressure is found when figuring the maximum explosion pressure at 450 pounds. When the engine is turning very slowly, the maximum load sustained by the pin will be the total pressure on the piston head caused by the explosion. At high speed the load will not be as great on account of the inertia of the piston. If we assume that 2500 pounds per square inch of projected area is the maximum pressure, and if l is the total length of bearing d the diameter of the piston pin, b the engine bore, and P (about 450 lb.) the maximum pressure on the piston head in pounds per square inch, we have the formula $2500ld = .7854b^2P$, from which $d = \frac{.7854b^2 \times P}{2500l}$. The piston pin bearing withstands only a rocking motion and therefore a high bearing pressure is permissible. On the other hand, high bearing pressures render the use of babbitt metal impractical on account of their softness. For this reason piston pin bearings are made of bronze, or the pin rocks directly in the piston; this is more frequently done with aluminum pistons, *i.e.*, the piston bosses form the bearings without any bushings, and it is found that they give fairly good satisfaction. Other manufacturers employ bushings. The piston pin is usually made of material having a high tensile strength, such as high carbon steel, nickel steel or nickel chrome steel, and in order to save weight, it is made hollow; the inside diameter of the pin being from one-half to three-fourths the outside diameter (depending upon the grade of steel used) is found to be of ample strength, when the diameter of the pin is calculated by the formula given before. The length of the bearing is usually from .60 to .65b when the pin rocks in the piston and from .45 to .55b when it rocks in the connecting rod. The diameter of the piston bosses when the pin is fastened in the bosses, or when the pin rocks in the piston without bronze bushings is made about 1.3 times the diameter d of the pin, tapering toward the piston wall where its diameter is 1.4d. When the piston

pin floats in the piston bosses and a bushing is used, the diameter of the latter is 1.5*d*, tapering toward the piston wall to about 1.6*d*.

It might be mentioned that with the large number of trucks used during the war in Europe, where defects occurred with pins, in numerous cases floating piston pins were substituted for the ordinary pins which gave trouble and it was found that this eliminated the said defects. That is to say, the pin was not locked in either the piston or the connecting rod but was free to rock in both. To prevent the ends of the pin (which is of steel, and thus harder than the cylinder material) from rubbing the cylinder wall, ordinary cast-iron stoppers are inserted in the piston, at the ends of the pin.

As to location of piston pin, a good plan, corresponding very closely to average practice, is to place it midway of the actual bearing surface length of the piston, *i.e.*, deduct the space actually occupied by the piston ring grooves and other grooves when figuring the length of bearing surface. Theoretically, it should be slightly nearer the piston head on account of the turning moment about the wrist pin created by the friction between the piston surface and the cylinder wall. This turning moment will of course be greatest during the power stroke.

In the piston of the Class B Engine for instance, which is 6⅛″ long, the actual bearing surface above the pin (after deducting all the grooves and the slant on the top) is 2⅜″; the actual bearing surface below the pin is (2⅝″ minus the two oil grooves of ⅛″ each) also 2⅜″.

Many designers disregard the piston space between the rings as wearing surface, when figuring the correct location of pin, but take the turning moment into consideration. Mr. Diamond in the S. A. E. Transaction (1916, Part I), gives the following formula for the correct pin location: If L is the total piston length, S the piston space form the head to the last piston ring groove, as shown in Fig. 72 and 72*a*, then K, the distance between the center of the pin and the outside of the head is,

$$K = \frac{L + S}{2} - \frac{cD}{2},$$

where c is the coefficient of friction, and D the diameter of the piston.

The coefficient of friction for aluminum alloy can be taken at .008, and for cast iron .014. To work out an example: let us say

the piston length is 5 inches, D is 4 inches, and the distance marked S on the drawing is $1\frac{1}{4}$ inches. Then the distance between the center of the wrist pin and the top of the piston head will be: $K = \frac{5 + 1.25}{2} - \frac{.008 \times 5}{2} =$ approximately $3\frac{1}{8}''$ from the piston head.

The last two figures show an example of alloy piston design largely taken from Mr. Diamond's illustration. Thickness of piston head $t = \frac{D}{16}$, (D = piston diameter).

$$A = t - \tfrac{1}{16},$$

$$u = \frac{D}{3},$$

$$O = .375D \text{ to } .35D \text{ (pin rocking in piston)},$$

$$d = \frac{D}{4} - \tfrac{1}{16},\ d' = \tfrac{3}{8}D - \tfrac{1}{16},\ d'' = \tfrac{3}{8}D,$$

$$G = \tfrac{15}{16}D - \tfrac{1}{16},\ H = \tfrac{15}{16}D.$$

It may be mentioned that in the design of trucks, where the engine speed is not very high, sometimes the diameter of the piston pin is slightly larger than that derived from the formula given before, while in high-speed engines it is sometimes slightly less.

CHAPTER X

CONNECTING-RODS

Connecting-rods transmit the power impulses from the piston to the crank shaft and should be made as light as possible, consistent with strength and rigidity. The main stress on a connecting-rod is that of compression; there is an additional stress tending to bend the rod, which is due to the transverse acceleration of its mass, called the whipping effect, and the stress caused by the inertia forces parallel to the axis of the cylinder.

CONNECTING-ROD LENGTH AND STRENGTH

The longer the connecting-rod, the lesser is the friction due to side thrust between piston and cylinder wall. On the other hand, a short connecting-rod will be lighter, thus having a reduced tendency to cause vibration, and for a given sectional area it will be able to withstand greater stresses.

Connecting-rods vary in length or in their ratio to the length of stroke, the average length being about 2.1 times the stroke. In high speed touring car engines the ratio will sometimes be 2 or even less, since a shorter stroke means a lighter reciprocating weight, and a lower, and thus a lighter engine. For truck engines the ratio is sometimes made 2.5 or even more. In the class B engine the length is 2.21 times the stroke, which is slightly longer than the average.

The section of the rod is usually I shaped, drop-forged from nickel steel (for inst. No. 2330 S. A. E. steel). At times when the saving in cost is very important, it is made of carbon steel (like No. 1035 S. A. E. steel).

Figure 77 shows the working drawings of the connecting-rod of the Class B Military truck engine. Note from the section, the usual draft of 7° given to the flanges, that is to say the ends of the web are slightly tapered, while the corners are rounded. Figures 78 and 79 show the proportion between the thickness of the web and the width and the height of the I-beam section, and rod lengths for engines made for touring cars and trucks. For touring cars (Fig. 78) the height of the connecting-rod as seen averages

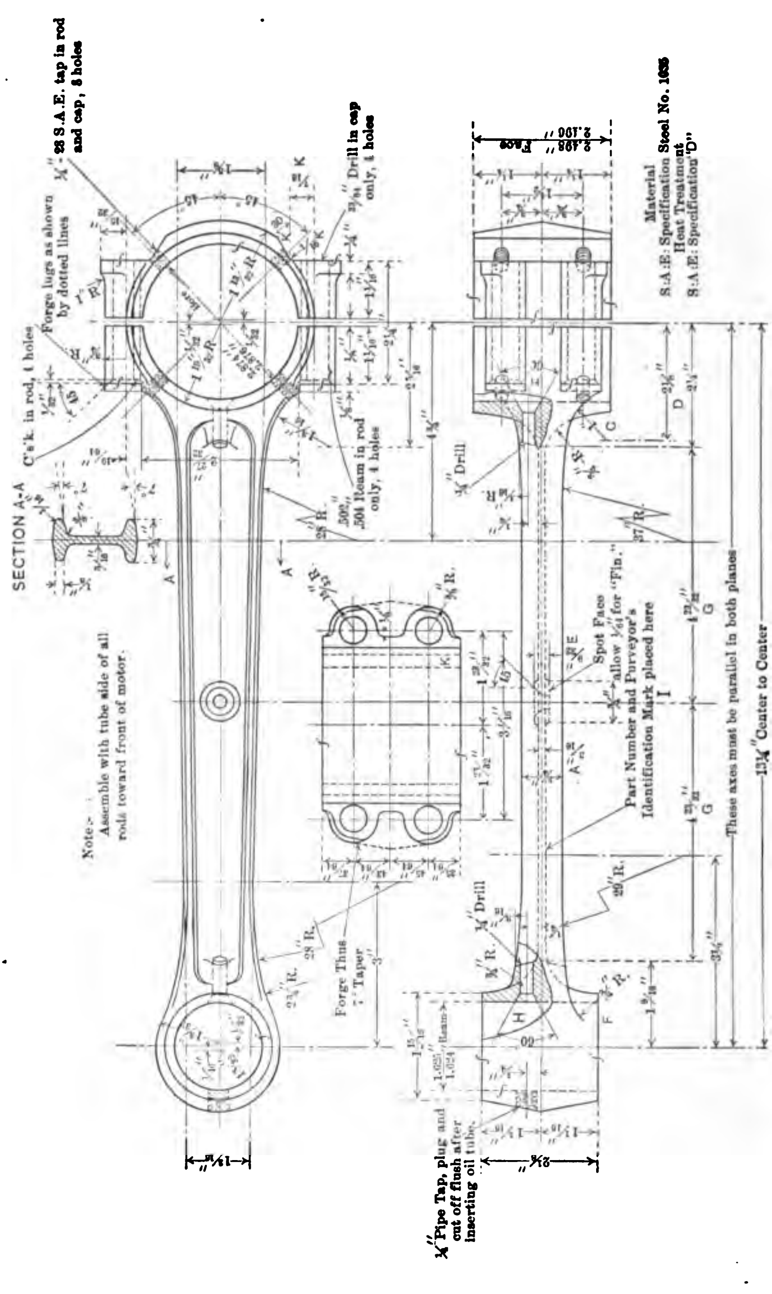


FIG. 77.—Connecting-rod of Class B, U. S. A. military truck engine (4¾" × 6").

6*t*, where *t* is the thickness of the web, while for trucks (Fig. 79) the height is 7*t*. The width is about 4*t* for both touring car and truck engines. It may be pointed out that this proportion varies considerably in the motors manufactured, but from a large number examined it is found that this figure represents good average practice. There seems to be no uniformity or rule among manufacturers for determining the actual strength of the rod; they are made considerably stronger than is necessary, in order to eliminate vibration. The curves plotted in the last two figures show the thicknesses of the web to be used on various lengths of rods and various engine bores.

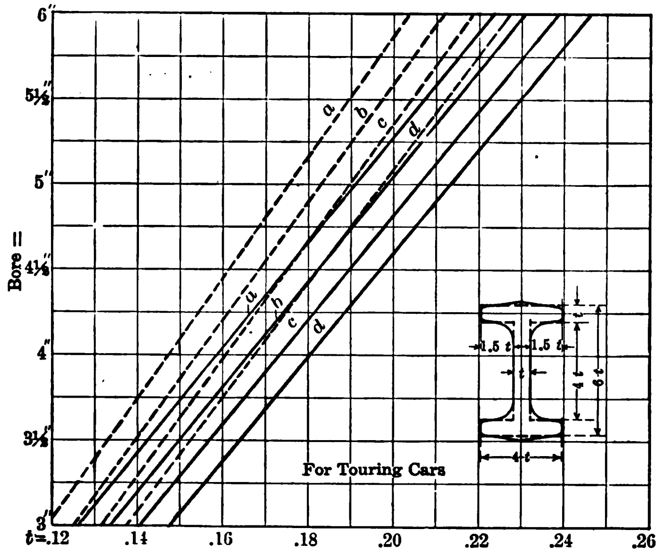


FIG. 78.—Dimensions of I-section connecting-rods for touring cars.

In figuring the stresses it is customary to use a factor of safety of 4, that is to say, in the nickel steel before mentioned the allowable working stress may be taken at $\frac{110,000}{4} = 27,500$ lb. per square inch; with the carbon steel mentioned it would be 21,250. The curves for both the carbon steel and the nickel steel have been plotted with these values.

From the curves the actual dimensions and thickness of the web can be seen at a glance without the use of mathematics. The thickness of the web and the proportions of the I beam have

been determined partly by the formula given by Mr. Heldt[1] and partly by actual designs of connecting-rods placed at the author's disposal by a number of automobile and truck engine manufacturers.

In the figures the dotted lines give the web thicknesses when nickel steel is used, and the solid lines for carbon steel. The letters *a*, *b*, *c*, and *d*, give the web thicknesses *t* for the following connecting-rod lengths:

a for rod length of 8″
b for rod length of 10″
c for rod length of 12″
d for rod length of 14″

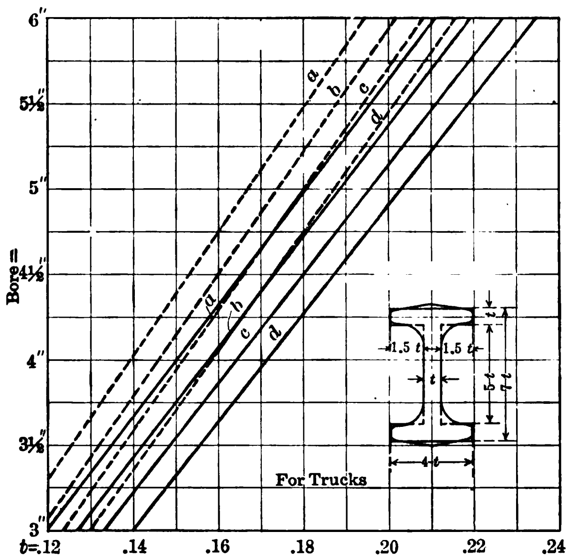


FIG. 79.—Dimensions of I-section connecting-rods for trucks.

Values for any other rod length can easily be determined from these curves. For instance say it is desired to find the web thickness for a rod made of carbon steel, a length of 10¼ inches, and a bore of 3½ inches, which are the dimensions of the rod shown in Fig. 70. Draw an imaginary line parallel to *b* and *c* and one-eighth their distance from *b* and we find the value for $t = .152$, approximately 5⁄32 inch.

In checking the value of *t* for the Class B motor we find $t = .195$, which is close to 3⁄16 inch.

[1] The Gasoline Automobile, by P. M. Heldt.

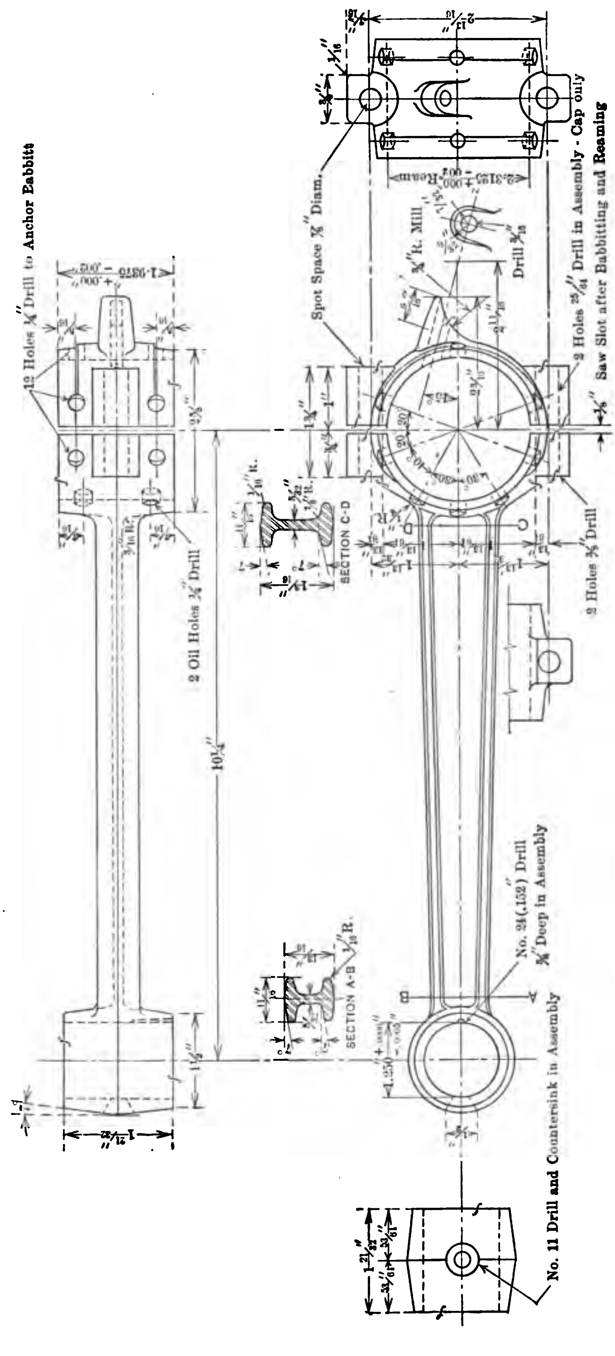


FIG. 80.—Connecting-rod of the Haynes light-six motor (3½" × 5").

In designing connecting-rods it should be remembered that the thickness of the web and the width of the rod are usually made uniform during the entire rod length, but the height of the section is made slightly greater near the lower end, and slightly smaller near the upper end. The dimensions of $6t$ and $7t$ given in figures represent the height in the center of the rod.

Figure 80 shows detail drawings of the Haynes Light Six Motor previously referred to, having a bore and stroke of $3\frac{1}{4}'' \times 5''$.

Figure 81 shows working drawings of the rod of the Continental Motor, Model 9*N*, six-cylinder, $3\frac{1}{2}'' \times 5\frac{1}{4}''$ motor. Note that in this case the lower end of the rod is slightly offset with respect to the upper end. By doing this, more room is obtainable for the center crankshaft bearings without lengthening the engine or the crankshaft. In this engine all the cylinders are cast in one block (see Figs. 39 to 39*e*). Examine also the crankshaft for this engine figure 87.

The stresses due to compression only can easily be determined and are the total pressure P on the piston head; $P = pa$, where p is the pressure per square inch and A the area of the piston head.

The total stress S on the rod per unit area would be:

$$S = \frac{P}{a} = \frac{pA}{a} \text{ where } a \text{ is the area of the connecting-rod.}$$

Looking at Fig. 78 for instance the area of the I rod is $6t \times 4t - 4t \times 3t = 12t^2$. As an example take a rod suitable for an engine of 4″ bore and length of rod 14″. We find in such an engine $t = .18$ (about $\frac{3}{16}''$) when carbon steel is used. The area of the rod therefore will be, $12t^2 = 12 \times (.18)^2 = .388$ square inch approximately.

The maximum compression stress on the rod when the maximum explosion pressure is taken at 400 lb. per square inch (at the beginning of the stroke when the rod is in line with the piston) would be:

$$S = \frac{pA}{a} = \frac{400 \times .7854 \times 4^2}{.388} = \frac{5020}{.388} = 12{,}890 \text{ lb. per square}$$

inch of area of the rod. (See Chapter XII, Inertia Forces.)

The strength of a rod depends however on its length as well as on its area, and when we consider the length and the bending moments, the stresses on the rod are enormous, and therefore the sections given in the curves are recommended.

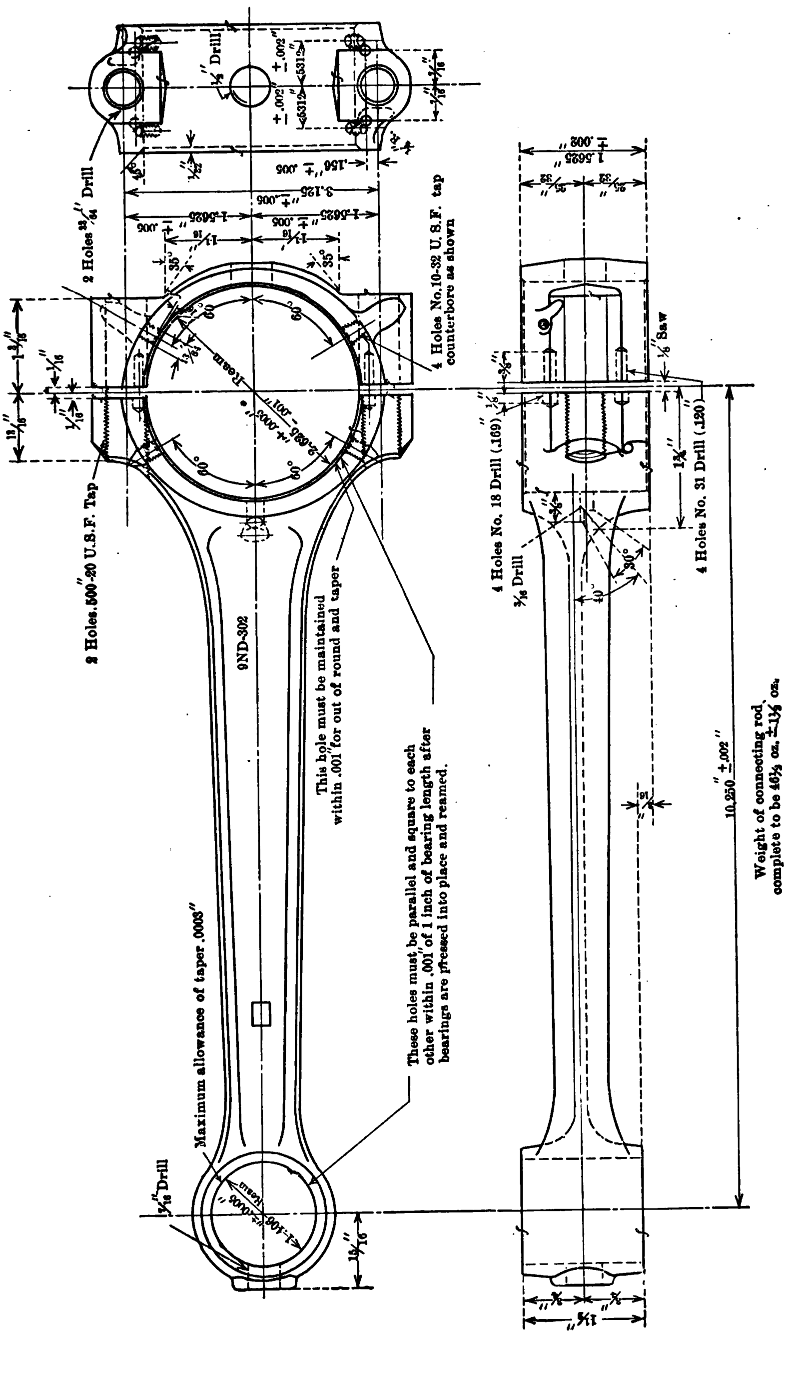


FIG. 81.—Connecting-rod of Continental motor, Model 9N, 6-cylinder ($3\frac{1}{2}'' \times 5\frac{1}{4}''$).

PISTON-PIN BEARINGS AND CONNECTING-ROD HUBS

The entire force of the explosion is transmitted from the piston to the connecting-rod through the piston pin, and therefore the piston-pin bearing has to withstand the entire pressure. The thickness of the bearing metal varies from about .125 d (d being the diameter of the piston pin) for small engines, to .085 d for large motors. The top end of the rod, which holds this bearing when the pin rocks in the rod, must be made of sufficient strength; on the other hand, increasing the weight means increasing the

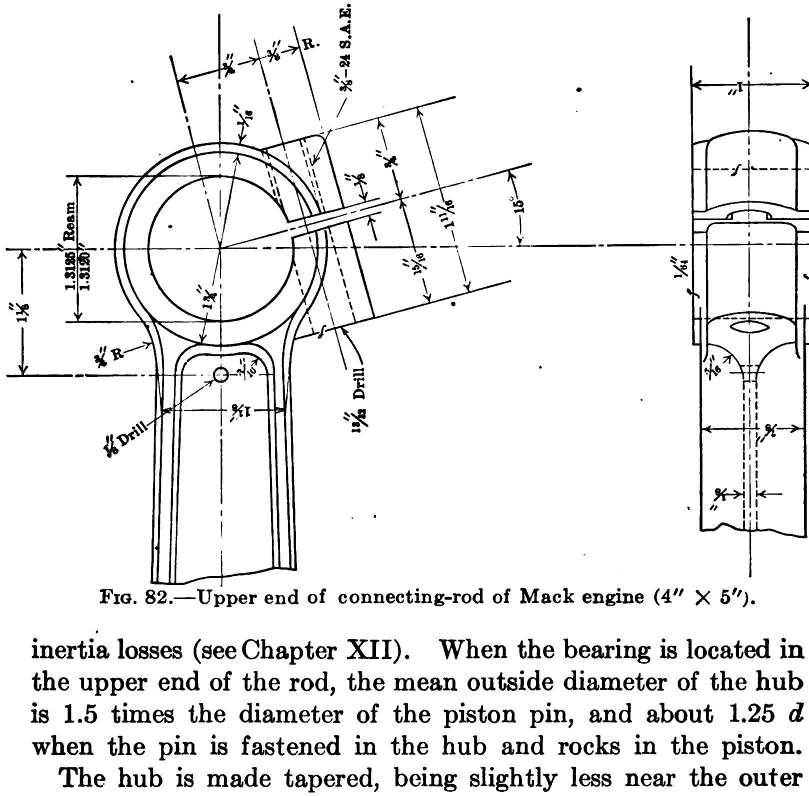


FIG. 82.—Upper end of connecting-rod of Mack engine (4″ × 5″).

inertia losses (see Chapter XII). When the bearing is located in the upper end of the rod, the mean outside diameter of the hub is 1.5 times the diameter of the piston pin, and about 1.25 d when the pin is fastened in the hub and rocks in the piston.

The hub is made tapered, being slightly less near the outer edge and slightly more in the center, see Figures 77 to 82.

To allow for any slight deviation in the cylinder position with respect to the crank-pin, it is necessary that the length of the hub be from 1/8 to 3/16 inch shorter than the distance between the piston bosses, and as there is no tendency for the rod to move sideways, this clearance will not cause any knock. The top of

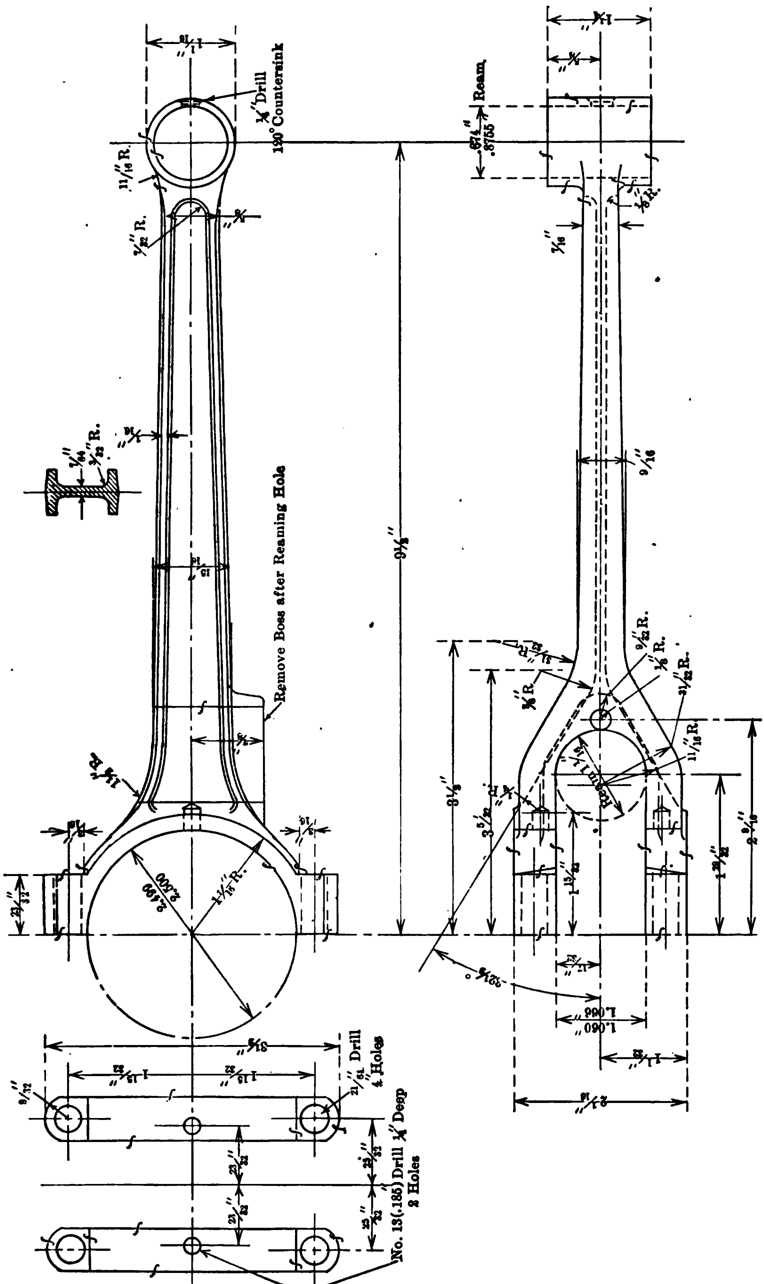


FIG. 83.—Connecting-rod (fork) of the Northway 8-cylinder motor ($2\frac{7}{8}'' \times 4\frac{3}{4}''$).

the hub is frequently provided with a hole, tapered, see Fig. 80, or with an oil catcher for particles of oil thrown out by the bearings (which fill the whole crankcase in the form of a mist), or thrown up by the splash, to lubricate the pin through the bushing. When pressure feed is used for lubricating the upper bearing in the rod provision must be made in the rod for attaching a pipe thereto as seen in Fig. 77. Fig. 82 shows the top of the rod provided with clamping means for preventing the piston-pin

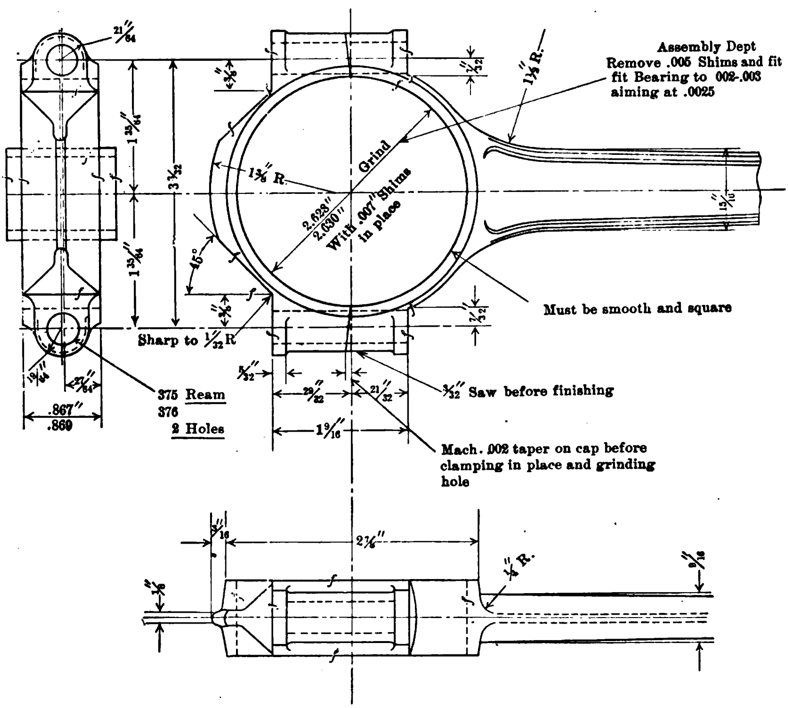


FIG. 83a.—Connecting-rod (blade) of the Northway motor.

from shifting. As a rule, when the pin rocks in the rod, its bearing bushing is held fast by making it a press fit, and in addition sometimes a pin or screw may be used, or it may be clamped tight by the rod similarly as shown in Fig. 82. Since the pressure per square inch is high, a hard bearing metal, like hard bronze or hardened steel must be employed.

The hub at the lower end of the rod, called the connecting-rod head, is made considerably larger in diameter than the upper end

since it contains the crank-pin bearing, and it is made with a detachable cap and split bushing, as seen from the figures. It is found more economical and more efficient, however, to drop-forge the rod with the cap in one piece, and saw it off after the machining operations are completed.

Laminated shims are used to make up for the saw cut and also for purposes of adjustment, when the bearings are worn or when they have to be scraped.

The thickness of the crank-pin bearing bushing, if made of babbitt lined bronze, for small engines it should not be made less than about 1/8 inch, as thinner bearings of this kind are more costly to manufacture. For larger engines they are made about 1/10 to 1/12 the diameter of the crank-pin. The author knows special cases where the bearing was made much thinner, probably for two reasons—to lighten the lower end of the connecting-rod (this would result in a reduced bearing pressure due to a reduced centrifugal force, see Chapter XII) and to give more clearance between the connecting-rod head and the crankcase. But as mentioned before, such thin bearings are more difficult to produce. The size of the studs holding down the cap will vary from 3/8 inch to 9/16 inch in motors having a bore of from 3½ inches to 5½ inches respectively. In order to make the studs light, nickel steel bolts are largely used. While four bolts are preferable it is difficult to drop-forge the connecting-rod head with lugs for four bolts and obtain lightness at the same time. For this reason two bolts are usually employed, for the smaller bore engines, and four bolts when the bore is 4½ inches or more, or when the motor is intended for special severe service.

In the U. S. Class B engine four bolts are used (see Fig. 77), while in Fig. 81, only two are employed.

In eight-cylinder engines where one-half the number of cylinders form an angle of 90° with the other half, or in twelves where they form an angle of 60°, two connecting rods are used on each crank-pin. Sometimes as in the Packard twelve and the King eight-cylinder engine the heads of all the rods are identical and are placed side by side on the crank-pin (see Fig. 104). At other times as in the National twelve and the Northway eight cylinder (Fig. 83 and 83*a*), one connecting-rod is forked, the head of the other, having a straight blade, is placed between the forked ends. All the dimensions are given on the drawings which should prove very instructive to students.

CHAPTER XI[1]

CRANKSHAFTS. ENGINE BALANCING

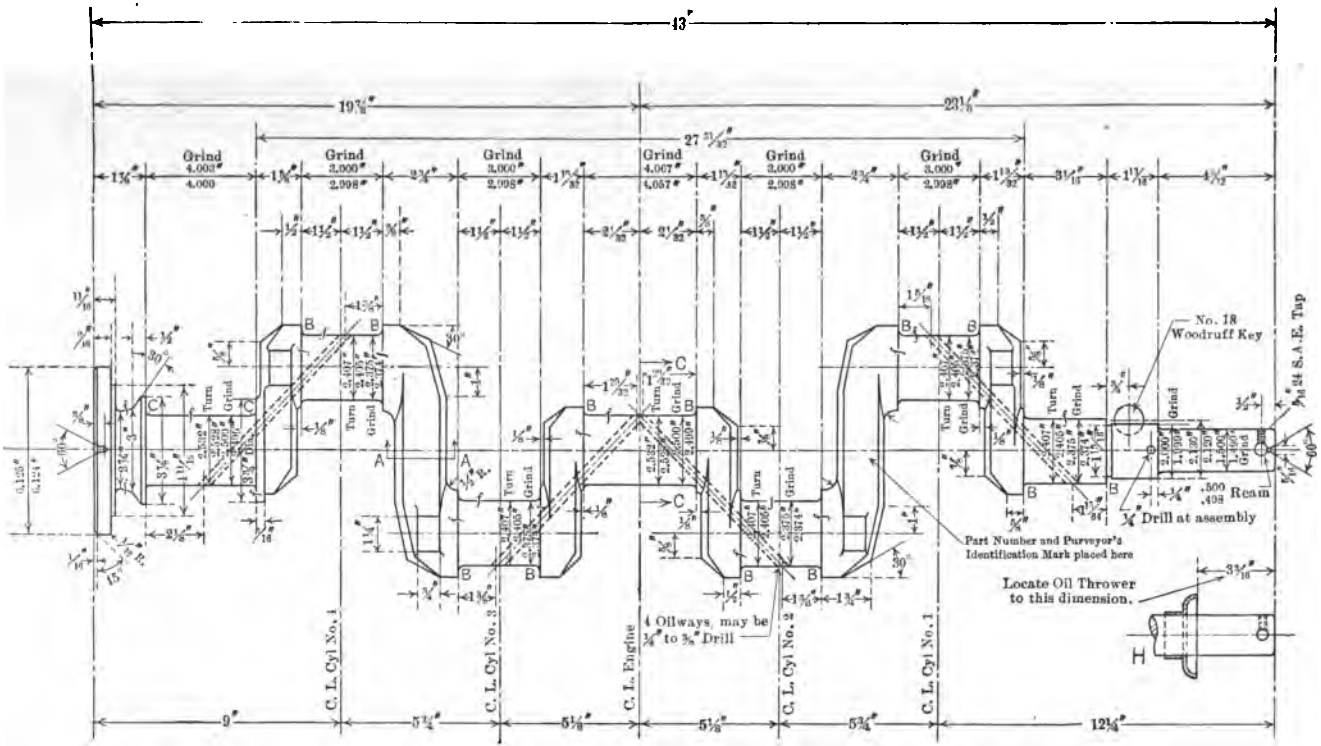
Internal combustion engines, as used in motor vehicle and airplane engines, where the speed at times reaches 2500 revolutions (or more) per minute, must have crankshafts sufficiently strong to withstand the various stresses at this high speed. The three most important stresses are those due to the pressure arising from the explosion, those due to the inertia effect of the reciprocating parts (this usually includes the weight of the piston, piston pin and upper half of the connecting-rod), and those due to the centrifugal force. The crankshaft is a comparatively expensive part of the engine, requiring very accurate machine work; unless made of proper dimensions and suitable material, it will give way.

The materials most extensively used for crankshafts are medium carbon steel or nickel steel; these shafts are drop forged, but occasionally, for high priced cars, they are made out of solid billets. (See the chapter on materials for the composition of the steel used.) The carbon steel used for this purpose (like No. 1045 S. A. E. steel) has a tensile strength of 110,000 lb. per square inch, and an elastic limit of 75,000 lb. per square inch. The nickel steel (like No. 2330 S. A. E. steel) has a tensile strength of 140,000 lb. per square inch.

Occasionally other steels, like chrome nickel steel, and chrome vanadium, are used. Chrome nickel steel, for instance, has a greater strength than nickel steel, but it is very hard to machine, and for this reason is not employed so frequently.

One of the main considerations in a crankshaft, is stiffness or rigidity, and this depends upon the coefficient of elasticity or the modulus of elasticity of the material, for this reason, the strength of the various materials does not much affect the size of a crankshaft, since the coefficient of elasticity (the modulus of elasticity) for all the different grades of steel is practically the same, about 30,000,000. (See Modulus of Elasticity, Chapter XXVII.)

[1] The formulæ in this chapter and some of the remarks have been taken from "The Gasoline Automobile" by P. M. Heldt by permission of the author.



Physical Properties After Treatment

Tensile Strength	110,000 lbs. per sq. in.
Elastic Limit	75,000 lbs. per sq. in.
Reduction in Area	45 %
Elongation in 2	17.5 %
Scleroscope	38 to 42

S. A. E. steel No. 1045 D. H. Heat treatment H.

Note:- B. Fillets are ⅛" radius unless otherwise noted
Shaft to be put in standing and running balance
Center main pin not to run out more than .002
Distance from rear bearing to any pin to be within ± 1/64"
Crank pins to line up with main pins within .001" per inch
All pins to be straight and round within .0005"
7° Draft Angle

FIG. 84.—Three-bearing crankshaft of Class B Military Truck engine (4¾" × 6").

The parts of a crankshaft are:

The crank-pins, to which the connecting-rod is attached;

The crank-arms, or cheeks, which are at each side of the crank-pins;

The crank-journals, which revolve in the main bearings;

The driving ends.

Occasionally, instead of being forged, the crankshaft is built up, and in some small engines abroad flywheels are enclosed in

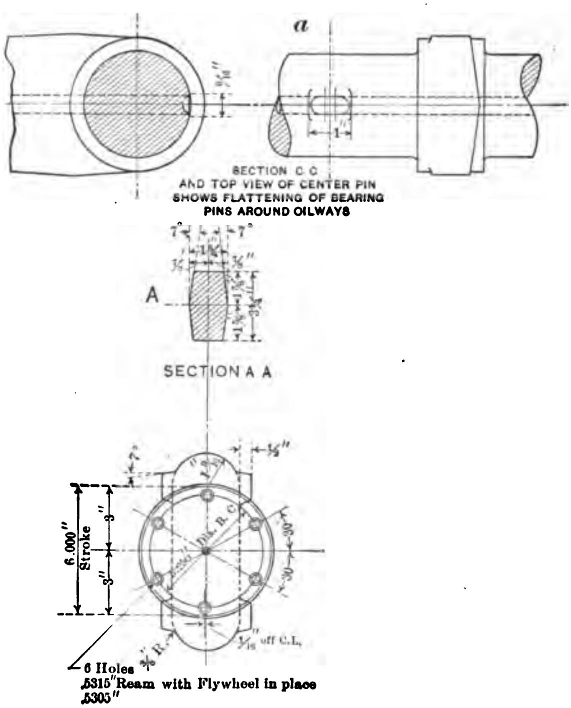


FIG. 84a.

the crank-case, that is to say, the crank-pins and crank-journals being bolted to rings forming the fly-wheels. Another reason for occasionally using built-up crankshafts is, that if it is desired to use ball bearings on the journals, with the integral crankshaft, the ball-bearing race would have to be of such a large diameter

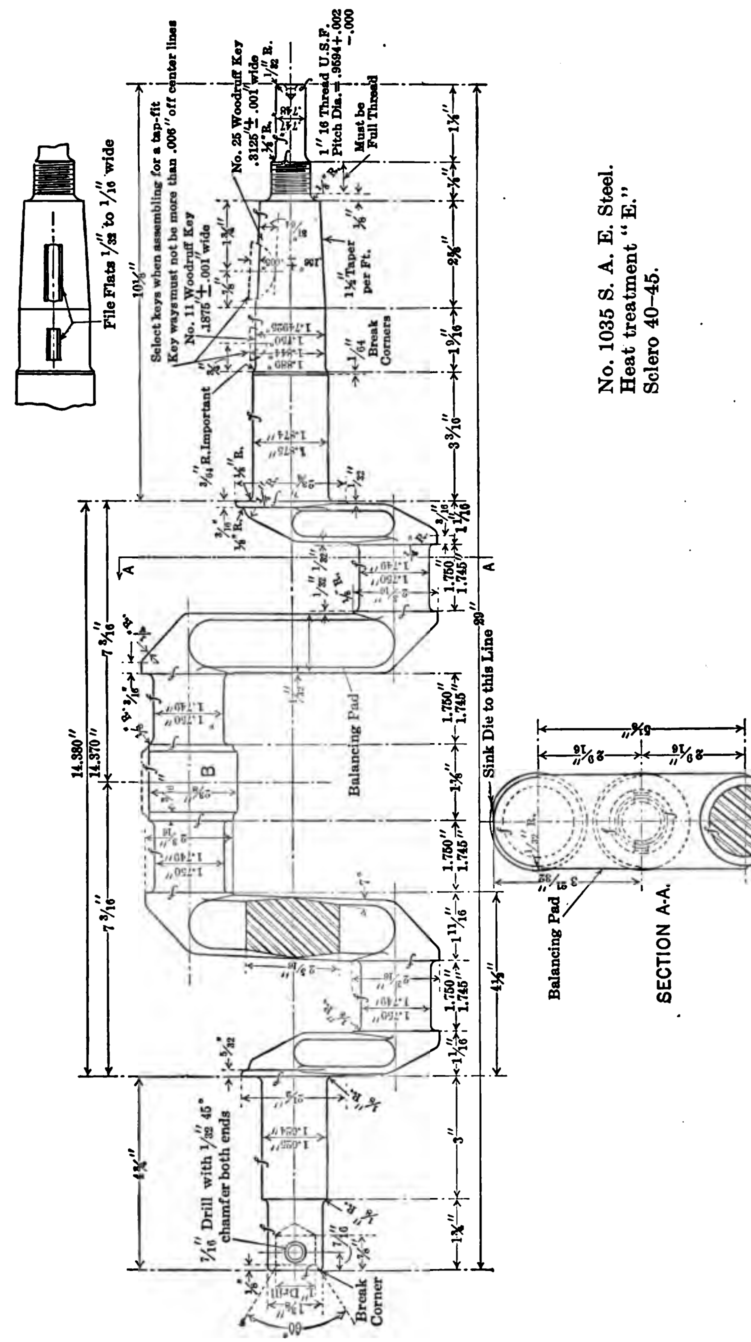


FIG. 85.—Two-bearing crankshaft of the Briscoe motor ($3\frac{3}{16}'' \times 5\frac{1}{8}''$).

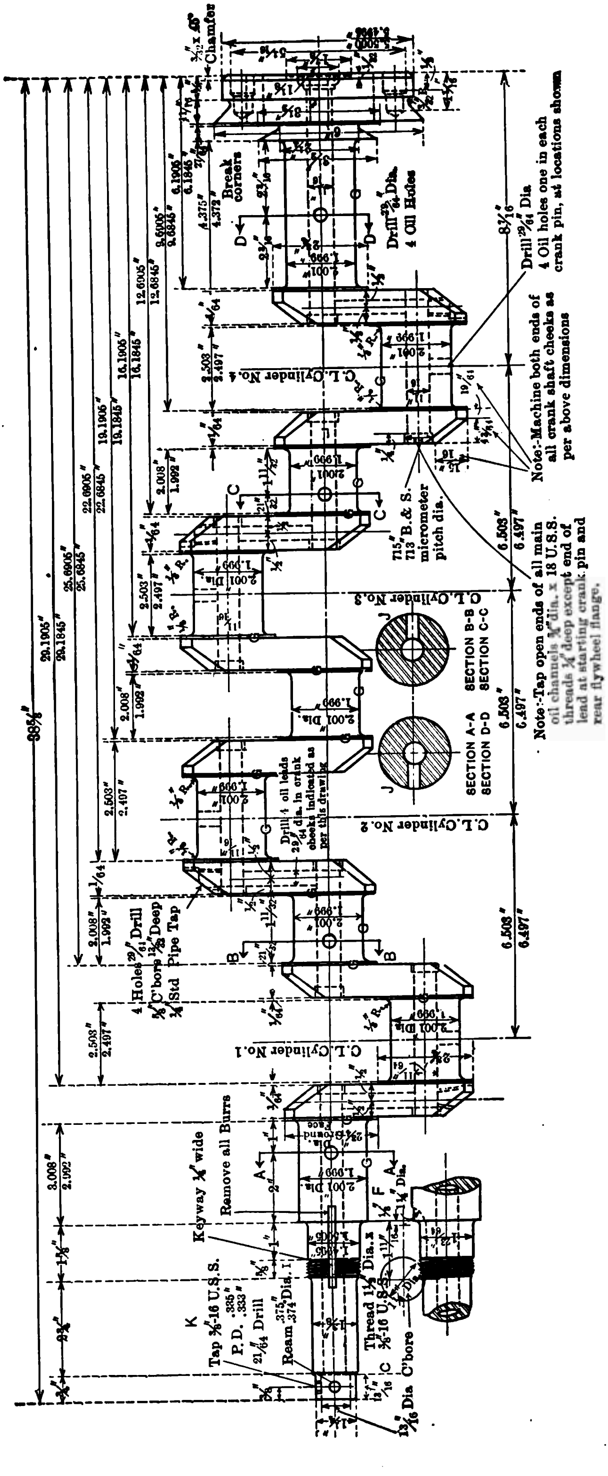


FIG. 86.—Five-bearing crankshaft of the Hercules, Series M motor (bore from 4" to $4\frac{1}{2}$", stroke $5\frac{1}{2}$").

as to be slipped over the crank-arms; by using a built-up crankshaft, the ball-bearing race need not be larger than the ordinary crank-journal diameter.

Figures 84 and 85 show the two common types of crankshafts used on four-cylinder engines. Figure 84 is a detail drawing of a crankshaft (U. S. Class B Military Truck Engine), having three bearings, and Fig. 85, the crankshaft of the Briscoe Motor with two bearings. Figure 86 is the crankshaft (five-bearing, four-cylinder) used on the Hercules Series M, Heavy Duty Motors for a bore varying from 4 inches to 4½ inches, with a stroke of

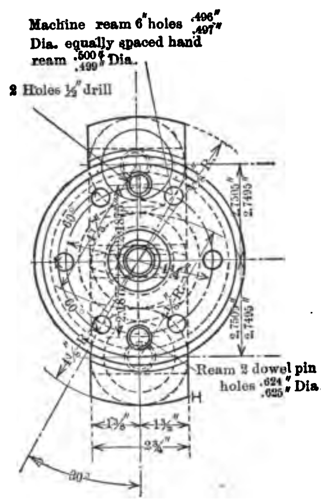


FIG. 86*a*.—End view of Fig. 86.

5½ inches. Here, as is noted, five bearings are used, one between each of the cylinders. In all working drawings wherever two dimensions appear (one set of figures underneath the other), it indicates the tolerance, *i. e.*, the maximum permissible variation in the actual dimension of the finished work. Figure 87 shows the three-bearing, six-cylinder crankshaft used in the Continental 9N motor referred to. Occasionally four bearings are used in six-cylinder crankshafts and sometimes seven bearings, one between each cylinder.

The crank-pin with its two crank-arms is often called a "throw," and it can be seen that sometimes one throw is used between two bearings, at other times, two, three, or four. For

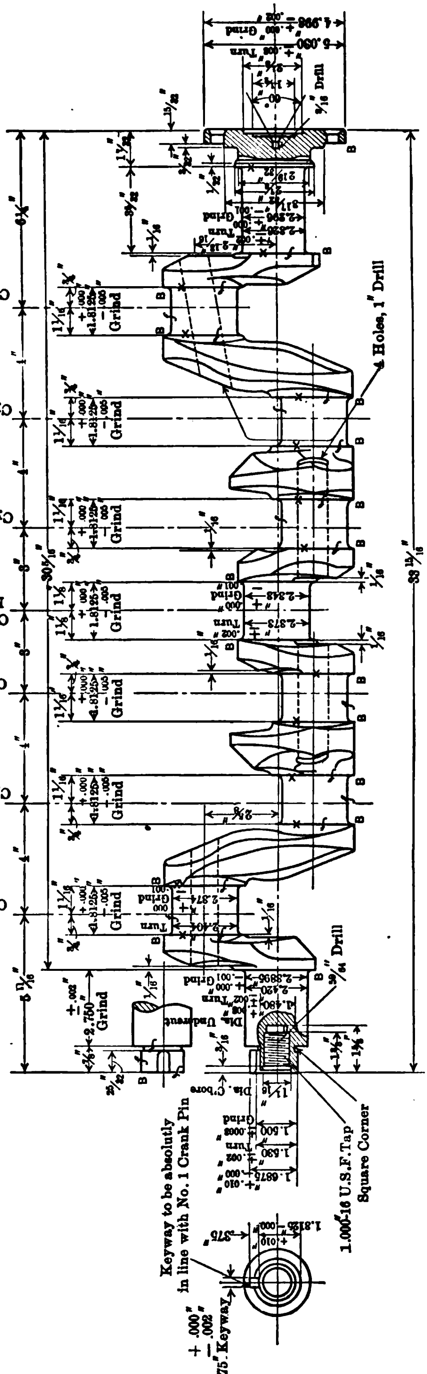


FIG. 87.—Three-bearing, six cylinder crankshaft of Continental motor Model 9N ($3\frac{1}{2}'' \times 5\frac{1}{4}''$).

four-cylinder engines the most common is the three-bearing crankshaft which of course has two throws between each pair of bearings, while for small four-cylinder engines the two-bearing shaft is frequently used, which has four throws between the bearings. Naturally, when there are several throws between the bearings the pins and arms must be of larger dimensions to withstand the bending stresses; for this reason, we may sometimes find a crankshaft used on larger engines having smaller dimensions than one used on smaller engines. In a four-cylinder engine the cranks are always in one plane, that is to say, the position of two of the throws are 180° apart from the other two throws. In such a shaft the weight of the throws on one side would balance those on the other side, but its individual throws are unbalanced when

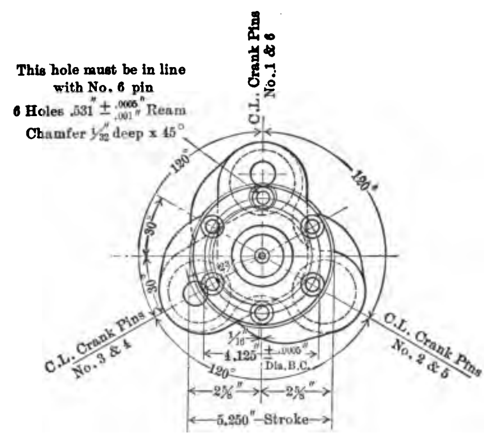


FIG. 87*a*.—End view of Fig. 87.

the explosion takes place in any one cylinder. As the crankshaft is to a certain extent flexible, the centrifugal force acting on the throws tends to produce a pressure on the crank-journals. To overcome this, a number of manufacturers counter-balance the crankshaft by means of balance weights opposite each short crank-arm. Of late the practice has not gained many adherents, but with the increase in engine speed an unbalanced crankshaft is naturally a serious matter, producing vibration of the engine and distortion of the shaft.

Lately some firms have used vibration dampers such as introduced by Lanchester of England, and used in the Packard Twin-Six engine.

Frequently designers dimension the crank-pin diameter proportionally to the cylinder bore, as the pressure arising from the explosion is considered the main factor in determining the crankshaft strength. On account of the high piston speed, however, the inertia of the reciprocating parts is in many cases more important, and for this reason the stroke should also be taken into account, since the longer the stroke the greater will be the inertia effect. When an engine is rotating at high speed, at the beginning of the explosion stroke there is very little pressure on the crank-pin as the force of the explosion is substantially balanced by the force of the inertia of the piston and the upper end of the connecting-rod, the force of which, acting in an opposite direction, tends to resist the force of the explosion. Since the force from the explosion of the gases acts only during one stroke of the cycle

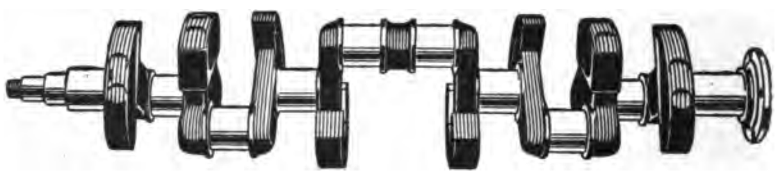


Fig. 88.—Crankshaft Hudson super-six.

while the inertia forces due to the reciprocating parts are acting during all the strokes, at high speeds the inertia forces have a greater influence upon the bearing load of the crank-pin than the explosion pressure. The inertia force is proportional to the weight of the reciprocating parts and the weight of the latter is proportional to the square of the bore and to the length of the stroke, that is to say, to the piston displacement; therefore the bearing surface of the crank-pin should be proportional to the piston displacement.

Bearing in mind the foregoing, the explosion pressure may be neglected in the calculation of crankshaft dimensions, especially as torsional rigidity and lateral stiffness have to be very great to avoid vibrations.

From a number of crankshafts of automobile motors, Mr. Heldt has compiled the following formulæ, which will be found to correspond to average practice. It should be noted that

there is a great difference in the dimensions of crankshafts used on similar engines manufactured by different manufacturers, and that since the inertia forces at high speeds are more important than the explosion pressure, in some instances truck engines will be found to have dimensions somewhat smaller than those determined by the formulæ, while in other cases engines used for touring cars have larger dimensions. This, however, does not hold good in all cases for the author knows a number of well-known truck engines which have dimensions quite considerably larger than those determined by the following formulæ.

Three-bearing Four-cylinder Crankshaft.—Consider first the most common type of crankshaft, that is to say, the three-bearing type for four-cylinder engines.

If d = the diameter of the crank-pin.

If L = the length of crank-pin bearing;

If d_1, d_2, d_3 = the diameters of the crank-journals from the front to the rear;

If L_1, L_2, L_3 = the length of the crank-journals, front to rear;

If C = the piston displacement of each cylinder in cubic inches;

If w = width of the crank-arms (widest portion);

If t_s = thickness of the short crank-arms;

If t_1 = thickness of long crank-arms.

The crank-pin diameter $d = \sqrt{\frac{C}{16}}$, and the length of the crank-pin journal $L = 1.25d$.

In the four-cylinder three-bearing type of crankshaft, all the crankshaft journals are as a rule made of the same diameter as the crank-pins, while the length of the different main journals are as follows:

$$L_1 \text{ and } L_2 = 1.25L \text{ each.}$$
$$L_3 = 1.75L.$$

The rear bearing which carries the weight of the flywheel in addition to other loads, has to be made longer.

Regarding the width of the crank-arms, this is made larger than the diameter of the crank-pin and the crank-journals to provide a bearing surface for end thrusts. The width of the crank-arms varies between 1.2d, and 1.58d. Sometimes a circular flange is provided, forged integral at both sides of the central journal or at both sides of the rear journal. In such cases the width of the arms can be 1.25d. On the other hand, if the

crank-arms are made of sufficient width to afford the required end thrust bearing surface, it is recommended to make $w = 1.4d$. The thicknesses of the crank-arms are different for the short and for the long arms. Where there is a bearing between each throw, all the crank-arms would be of the same length. However, when there are two throws between a pair of bearings, naturally, the center arm connecting one pin with the opposite pin is double the length of the short arms and must therefore be made stronger, for the bending moment is greater in the longer arm; therefore, the thickness for the smaller arm $t_s = .6\sqrt{\frac{d^3}{w}}$, and the thickness of the long arm $t_1 = .8\sqrt{\frac{d^3}{w}}$.

When forging the crankshaft, the draft allowed in the arms is usually about 7° as shown at A in the section of Fig. 84. Let us work out an example for a three-bearing four-cylinder crankshaft having a bore of 3½ inches, and stroke of 5 inches. In such a motor the piston displacement $C = 48.1$ cubic inches. In this case, the diameter of the crank-pin $d = \sqrt{\frac{48.1}{16}} = 1.74$ or approximately 1¾ inches. The length of the crank-pin journal or crank-pin bearing $L = 1.25 \times 1.74 = 2.18$ or approximately 2³⁄₁₆ inches. The length of the front and the middle main-crank-journals L_1 and $L_2 = 1.25 \times 2.18 = 2.7$ or approximately 2¾ inches. The rear main bearing $L_3 = 1\frac{3}{4} \times 2.18 = 3.82$ or approximately 3¾ inches, and the width of the crank-arms $w = 1.4 \times 1.74 = 2.43$ or approximately 2½ inches. The thickness of the short crank-arms $t_s = .6\sqrt{\frac{1.75^3}{2.5}} = .876$ or approximately ⅞ inch, and the thickness of the long crank-arms $t_1 = .8\sqrt{\frac{1.75^3}{2.5}} = 1.17$, approximately 1³⁄₁₆ inches.

The front end of all crankshafts must have sufficient length to provide for the driving pinions of the camshaft gear and the pin or ratchet sleeve which engages the starting-crank. There frequently is an oil guard or oil thrower provided on the front end between the ratchet and the cam-gear (see H, Fig. 84). The length of cam-gear hub is made about .6 times the crank-shaft diameter, and there is an additional length provided for the ratchet sleeve to slide on, as shown in the figures.

The thickness of the flange which is bolted to the flywheel may be made about $.062 \times D \times N$ where D is the cylinder bore

and N the number of cylinders. The outside diameter of the flange is usually equal to the stroke and there are six holes in it for the bolts by which it is fastened to the flywheel, while the diameter of the bolts is about the same as the width of the flange. Beyond the flange the crankshaft is frequently extended a short distance to form a stud or pilot for the clutch to rest on.

ENGINE BALANCING

Before proceding to give the dimensions for other types of crankshafts, it is thought advisable to speak about engine vibration and crankshaft balancing, as this is a very important subject in high-speed engines. There are various causes of vibration in an engine, some of which can be overcome to a great extent by the proper balancing of the crankshaft. The causes of vibration are:

Lack of crankshaft balance. (May be out of static balance or running balance. Can be effectively remedied.)

Distortion in crankshaft. (Can be overcome to some extent by balance weights.)

Reciprocating parts out of balance. (Can be effectively remedied.)

Lack of flywheel balance. (Can be remedied.)

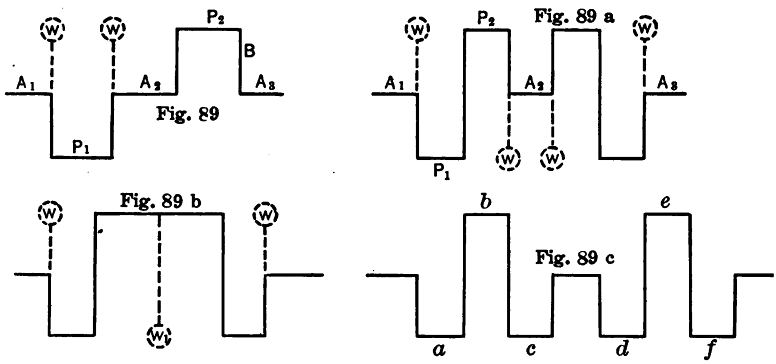
Torque reaction. (Cannot be remedied.)

Torque reaction is caused by the force acting against the cylinder wall when the piston is forced downward by the pressure of the explosion and exerts a force or turning moment upon the crankshaft. The force tending in an opposite direction, against the cylinder wall, is the reaction which is equal and opposite to the pressure exerted upon the crank-pin. As this torque reaction on the cylinder wall is intermittent, it will set up vibrations which cannot very well be overcome.

Figure 89[1] shows diagrammatically a two-throw crankshaft in which A is the axis or the journals, B the crank-arms, and P the crank-pins. In this type, if P_1 and P_2 are equal in weight, the crankshaft can easily be in what is termed static balance, that is to say, the weight of the pins P, on one side between bearings A_1 and A_2 will balance the weight of P_2 between A_2 and A_3, and this balance is secured when the axis of a mass passes through the center of the mass. In other words, when the crankshaft is placed on two level straight-edges, and will balance in any position, it is termed the "crankshaft is in static-balance." This, however, does not mean that the shaft is in running-balance, or in distortion-balance. In order to have the shaft in running-balance, the

[1] See A. P. Brush, Problem in High-Speed Engine Design. S. A. E. Transactions, 1916, Part II.

centrifugal couple between any two bearings should balance and neutralize each other, when the shaft is rotating; unless this is done, there is a tendency to throw the axis A out of alignment. In order to prevent undue stresses in the shaft and in the shaft bearings and thus prevent vibrations, it is necessary that the shaft be in perfect running balance between each two bearings, and this can be accomplished by adding counter-balance weights W as shown dotted in Fig. 89. Vibration means loss of power, for the energy which maintains the vibrations must be produced by the engine.



FIGS. 89, 89*a*, 89*b*, AND 89*c*.—Counter-balanced crankshafts.

Figure 89*a* is a three-bearing four-throw crankshaft, and there are two throws and two crank-pins between each two bearings and therefore this may be termed to be in running balance. Yet there is a tendency for the shaft to be distorted when explosion takes place in any one cylinder; this can be counter-balanced to a great extent by weights W as shown dotted.

Figure 89*b* shows a four-throw two-bearing crankshaft (as used on a number of small engines) with counter-weights shown, to obtain a complete balance.

It might be mentioned that there is always a certain amount of distortion in the shaft but by having counter weights as nearly as possible opposite each pin, its distortion is very largely reduced and hence there are less vibrations in such a shaft.

To secure the minimum tendency of distortion the counter-balance weights should be greater than that necessary to counter-balance or neutralize the centrifugal effects of the crank-pins and the crank-cheeks. This increase must be allowed on account

of the weight of the reciprocating parts. It is also evident that the fewer the number of bearings in a crankshaft the stronger must the crankshaft be made in order to reduce the strain and therefore the vibrations.

TWO-BEARING FOUR-CYLINDER CRANKSHAFTS

In two-bearing four-cylinder crankshafts as used on small bore high-speed engines, the load exerted upon the bearings when the shaft is not in distortion-balance, is greater than the load arising from the explosion pressure in the cylinders. This is the reason a great deal of trouble was encountered in the beginning with this type of crankshaft. If they are made sufficiently stiff (see Fig. 85), or if they are properly counter-balanced, they give fairly good satisfaction. In this crankshaft the center balance weight W_1 (see Fig. 89*b*) should be somewhat greater than twice the weight W of the end counter-balance weights, as the center crank-pin which carries two connecting-rods is longer than the two end pins combined. This difference in length, and hence in weight, must be taken into account when figuring the centrifugal force of the center counter-balance weight. Two-bearing four-cylinder crankshafts are ordinarily only employed on engines with a piston displacement of less than 200 cubic inches, and the crank-pin and crankshaft journals are as a rule made of the same diameter. Using the letters we employed in the former paper, that is to say,

If d = the diameter of the crank-pin

If L = the length of crank-pin bearing

If d_1 and d_2 = the diameters of the front and rear crank-journals.

In a two-bearing four-cylinder crankshaft, d_1 and d_2 are usually made of the same diameter as the crank-pin d. Therefore, in this case, d_1 and $d_2 = d$.

If L_1 and L_2 = the length of the front and the rear crank-journals,

If C = the piston displacement of each cylinder in cubic inches,

If w = the width of the crank-arms (widest portion),

If t_s = the thickness of the short crank-arms,

If t_1 = the thickness of long crank-arms, we have:

The diameter of the crank-pin $d = \sqrt{\frac{C}{12.5}}$.

The length of the crank-pin bearing $L = d$.

The front crank-journal $L_1 = 1.3L$.
The rear crank-journal $L_2 = 1.7L$.
The width of the crank-arms w (greatest width) $= 1.25d$.
The thickness of the short crank-arm $t_s = .55\sqrt{\frac{d^3}{w}}$, and
The thickness of the long arm $t_l = .8\sqrt{\frac{d^3}{w}}$.

Let us work out an example for an engine having a $3\frac{1}{2}$-inch bore and a 5-inch stroke. The piston displacement of such an engine will be $.7854 \times 3\frac{1}{2} \times 3\frac{1}{2} \times 5 = 48.1$ cubic inches per cylinder. In this case we have:

$$
\begin{aligned}
d &= \sqrt{\frac{48.1}{12.5}} - 1.96, \text{ approximately } 2'' \\
L &= d = 2'' \\
L_1 &= 1.3L = 2.6026, \text{ approximately } 2\tfrac{5}{8}'' \\
L_2 &= 1.7L = 3.4034, \text{ approximately } 3\tfrac{3}{8}'' \\
w &= 1.25d = 2.5025, \text{ approximately } 2\tfrac{1}{2}'' \\
t_s &= .55\sqrt{\frac{d^3}{w}} = .9845, \text{ approximately } 1'' \\
t_l &= .8\sqrt{\frac{d^3}{w}} = 1.43, \text{ approximately } 1\tfrac{7}{16}''.
\end{aligned}
$$

The four-throw two-bearing crankshaft shown in Fig. 85 is not symmetrical and is therefore not statically balanced. The two long arms balance each other, but the two short arms are both on the same side and they must be balanced by a block of metal B between the two middle crank-pins. To obtain static balance, this block of metal must be of sufficient diameter so that it balances the two short crank-arms. In addition, to obtain the best results, *i.e.*, to obtain distortion-balance, this type of crankshaft should be balanced by counter weights as shown in Fig. 89*b*, and as mentioned in the beginning of this paper, yet some manufacturers prefer not to use balance weights and instead make the shaft stiffer.

In this type of crankshaft the end thrust in both directions can be taken up on thrust collars on both sides of the rear main journal, similarly as shown at C, Fig. 84. When such provision is made to take up the thrust at certain points, a small clearance must be allowed at other points.

SIX-CYLINDER THREE-BEARING CRANKSHAFTS

The most popular for six-cylinder engines is the three-bearing type (Fig. 87), while the four-bearing type is also largely used. The seven-bearing six-cylinder shaft is also used in some cases but not so frequently. The three-bearing crankshaft is naturally the most suitable where the cylinders are cast in 3's; the four-bearing type where the cylinders are cast in pairs; and the seven-bearing where the cylinders are cast separately. Of course any of these three types of crankshafts may be used with any type cylinder, except that those mentioned are as a rule the most easily applicable. Let us first consider the three-bearing six-cylinder shaft. In this case, there are three throws between each two bearings, and therefore the crank-pins and arms must be very strong. Using the letters we employed before for denoting the different parts of the crankshaft, except that L_1 and L_2 and L_3 are the front journal, the middle, and end journal, respectively, we have the following formulæ:

$$d = \sqrt{\frac{C}{12}}$$

$$L = 1.06d$$

(d_1, d_2 and d_3, the diameters of the main journals, $= d$)

$$L_1 \text{ and } L_2 = 1.3L$$

$$L_3 = 1.75L$$

$$w = 1.25d$$

$$t_s = .55\sqrt{\frac{d^3}{w}}$$

$$t_1 = .8\sqrt{\frac{d^3}{w}}.$$

By working out an example for a $3\frac{1}{2} \times 5''$ motor as in the former example, we have a piston displacement C of 48.1 cubic inches. The crank-pin diameter d is 2″, the

Crank-pin length $L = 1.06d \doteq 2\frac{1}{8}''$ approximately, the
Diameter of the main journals d_1, d_2 and$_3$ $d = d = 2$ inches
L_1 and $L_2 = 1.3L$ approximately $2\frac{3}{4}''$, the
Length of the rear journal $L_3 = 1.75L$ approximately, $3\frac{3}{4}''$
$w = 1.25 \times 2'' = 2\frac{1}{2}$.

Thickness of short arm $t_s = .55\sqrt{\frac{2^3}{2.5}} = 1$ inch

$$t_1 = .8\sqrt{\frac{2^3}{2.5}} = 1\tfrac{7}{16}''.$$

In the six-cylinder three-bearing crankshaft there are three throws between each pair of bearings; the crank-pins in the center, *b* and *e* (see Fig. 89*c*), between each pair of bearings have a long arm on each side, while all other crank-pins have a long arm on one side and a short arm on the other. The adjacent crank-pins are always 120° apart, that is to say, *a*, *f*, are 120° apart from *b*, *e*, and from *c*, *d*. The two pins *c* and *d*, at the side of the center bearing are in one plane. The long arms, in order to se-

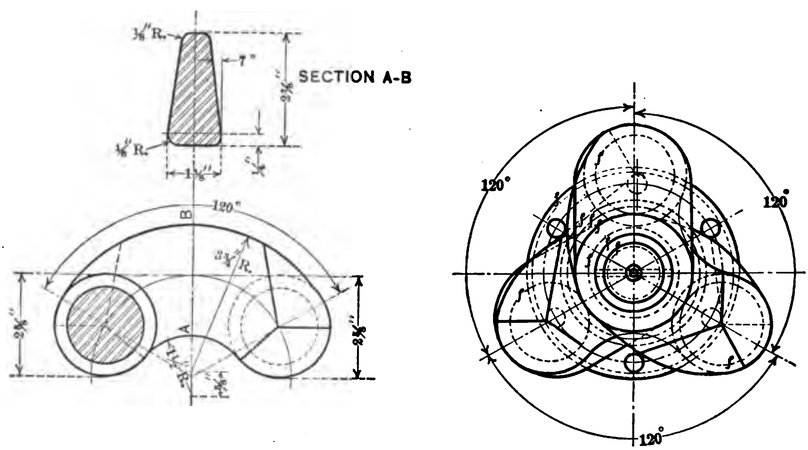


FIG. 90.—Section and end view of Haynes light-six motor crankshaft (3½″ × 5″).

cure balance, should run from one pin inward to the crankshaft axis and then outward to the adjacent crank-pin as shown in Fig. 90 instead of straight across from one crank-pin, to the other. This type of crankshaft is used principally where all the cylinders are cast en block or in two sets of three each. When they are all cast en block there must of course be some space left between the third and fourth cylinders to make room for the center crankshaft bearing, but by offsetting the connecting-rod as in Fig. 81 this space may be comparatively small.

SIX-CYLINDER FOUR-BEARING CRANKSHAFT

This type is especially applicable where the cylinders are cast in pairs as there is naturally more space between each pair of

cylinders and such space is required for the crankshaft journals when they are located between each adjacent pair of cylinders. As there are only two throws between each pair of bearings the crank-pins and arms can be made somewhat lighter than in the three-bearing six-cylinder crankshaft. The formulæ to find the proper dimensions are as follows:

Crank-pin diameter $d = \sqrt{\frac{C}{15}}$

Crank-pin length $L = 1.1d$,

Front journal $d_1 = \sqrt{\frac{C}{15}}$

d_2 and $d_3 = \sqrt{\frac{C}{13}}$

The rear bearing $d_4 = \sqrt{\frac{C}{12}}$. The reason for making the rear journals heavier is due to the increased torsional or turning effort (arising from the explosion pressure and inertia forces), since they vary continuously, and the flywheel prevents sudden changes in the motion of the cranks, therefore the crankshaft will have a tendency to twist under these variable torsion forces. It might be mentioned that every crankshaft has a certain definite period of torsion vibration, and if the engine speed happens to be such that the interval between the explosions is equal to the natural period of crankshaft vibration, very severe vibrations will be set up. For this reason the shaft must be made very stiff torsionally and in this respect that crankshaft journal bears most of the brunt which is subjected to the greatest torque.

Length of bearings L_1 L_2 $L_3 = L$, and the

Length of the rear bearing $L_4 = 1.6L$, the

Width of the arms $w = 1.35d$, the

Thickness of the short arm $t_s = .65\sqrt{\frac{d^3}{w}}$, the

Thickness of the long arm $t_l = .85\sqrt{\frac{d^3}{w}}$.

SIX-CYLINDER SEVEN-BEARING CRANKSHAFT

In a seven-bearing crankshaft, the distances between the bearings are very short, thus the bending moments are small, and therefore the crank-pins could be made of smaller diameter, but

as the shaft itself is very long, it is more liable to torsional vibration.

To secure adequate bearing surface without unnecessarily lengthening the shaft, the crank-arms should be finished on the sides and the journals should come right up to the sides of the arms, except for a small fillet.

Figure 88 is the four-bearing, six-cylinder, crankshaft of the Hudson super-six.

EIGHT-CYLINDER THREE-BEARING CRANKSHAFTS

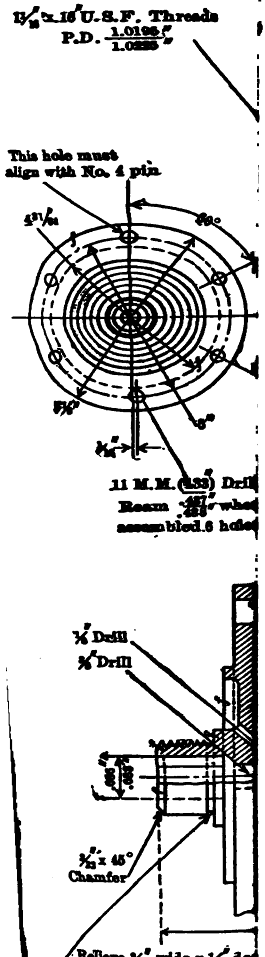
In eight-cylinder motors where two sets of cylinders are at an angle of 90°, any type of crankshaft suitable for four-cylinder motors may be employed, but the two- or three-bearing types are those generally used. In these motors two connecting-rods are placed on each crank-pin as was stated in the chapter on connecting-rods. The two methods prevalent for arranging the connecting-rod heads on the pins are either to fork one connecting-rod and clamp it to a bearing bushing with the other connecting-rod bearing on the outside of the bushing, or to have two narrow blade connecting-rod heads with bearings, placed side by side on the pins.

The crank-pins and the crank-journals are usually made of the same diameter as the journals.

EIGHT-CYLINDER TWO-BEARING CRANKSHAFT

The center bearing in a three-bearing shaft necessitates considerable space between the second and third cylinders, and for this reason a two-bearing shaft is often preferred by designers. In an eight-cylinder engine the bore is as a rule so much smaller than in a four, and the reciprocating parts so much lighter that the two-bearing shaft is more frequently used in the eight than in the four.

Figure 91 shows the balanced four-throw, eight-cylinder ($2\frac{7}{8} \times 4\frac{3}{4}$) two-bearing crank shaft of the Northway engine as used on the Oldsmobile. All the dimensions are given and need no further explanations. Attention is drawn to the balance weights, and the to size of the center weight compared to the two outside weights.



FIGS. 91 AND 91a.—T

TWELVE-CYLINDER THREE-BEARING CRANKSHAFT

Figure 92 shows detail drawing of the counter-balanced crank shaft of the three-bearing twelve-cylinder National engine.

Many designers instead of using the formulæ before given for determining crankshaft and crank-pin diameters use their experience derived from practice. Mr. D. Fergusson[1] mentions the following rule-of-thumb as a fairly good comparison with most of the successful medium-speed car and truck engines on the market, having stroke-bore ratios of from 1.3 to 1.6, and having reciprocating parts of medium weight:

In a four-cylinder, three-bearing crankshaft engine, the diameter of crank-journals and pins is made .5 of the cylinder diameter.

In a six-cylinder seven-bearing crankshaft engine, the diameter of the journals is made from .55 to .6 of the cylinder diameter.

In an eight-cylinder three-bearing crankshaft engine, the diameter of the journals and pins is made from .65 to .7 of the cylinder diameter.

In a twelve-cylinder three-bearing crankshaft engine, the diameter of the journals and pins is made from .75 to .8 of the cylinder diameter.

[1] S. A. E. Transaction, 1917, Part II, page 247.

CHAPTER XII

KINETIC FORCES

When a mass is set into motion, and when its speed is increased it requires the application of a force whose magnitude depends on the mass and on the rate of speed at which the change takes place. (The change of speed in unit time is called the acceleration.) In like manner when the speed of a moving mass is retarded, a certain force is necessary to cause the retardation. Such forces are called kinetic forces or inertia forces. The motion of the piston with the upper half of the connecting-rod is the chief source of the inertia forces in an internal combustion engine.

Another kinetic force is created when a mass revolves around an axis, in which case there is a tendency of the mass to move radially outward from the center (it tends to fly away from the axle), and this is called the centrifugal force. We have the formula:

Mass $M = \frac{W}{g}$; where W is the weight of the mass in pounds, and the acceleration due to gravity $g = 32.2$ feet per second, per second.

CENTRIFUGAL FORCES

The centrifugal force $CF = \frac{WV^2}{gR} = \frac{WV^2\,12}{32.2\,r}$, where V = the lineal velocity of the center of the revolving mass (the crank-pin) in feet per second.

R = radius at which the center of gravity of the mass revolves in feet.

r = radius at which the center of gravity of the mass revolves in inches.

For V we have the ordinary formula:

$V = 2\pi R \frac{N}{60}$, where N = the number of revolutions per minute,

and $\frac{N}{60}$ = the number of revolutions per second.

and $V^2 = \left(\frac{2\pi RN}{60}\right)\left(\frac{2\pi RN}{60}\right) = \frac{4\pi^2R^2N^2}{3600} = \frac{4\,(3.1416)^2}{3600}\,R^2N^2 =$

.010966 R^2N^2. Substituting this for V^2 in the formula before given, we have:

$$CF = \frac{WV^2}{gR} = \frac{W.010966R^2N^2}{32.2R} = \frac{.010966}{32.2}WRN^2 = .00034WRN^2.$$

For example in the Class B engine the lower end of the connecting-rod with the bearing weighs 5½ lb., the crank-radius is 3″ (.25 ft.); therefore at 1500 r.p.m. the centrifugal force $CF = .00034 \times 5.5 \times .25 \times 1500^2 = 1052$ lb. At 750 r.p.m. the $CF = 263$ lb.

INERTIA FORCES

The inertia force $IF = \frac{Wa}{g}$, where a = the acceleration of the mass in feet per second, per second. The acceleration of the reciprocating parts of an engine varies constantly since the piston comes to a stand-still when it reaches the end of its stroke and attains a maximum velocity between the inner and outer dead centers. It can be proven mathematically that for a given angle A of the crank arm the acceleration of the reciprocating parts in a motor is approximately as follows:

$$a = \frac{V^2\left(\cos A + \frac{r}{L}\cos 2A\right)}{R},$$ by substituting the value we found for V^2, we have:

$$a = \frac{.010966R^2N^2\left(\cos A + \frac{r}{L}\cos 2A\right)}{R}$$

$$= .010966RN^2\left(\cos A + \frac{r}{L}\cos 2A\right).$$

Where r = radius of crank in inches, and
L = length of connecting-rod in inches.

The inertia force $IF = \frac{Wa}{g}$, and substituting the value for a we have:

$$IF = \frac{W.010966RN^2\left(\cos A + \frac{r}{L}\cos 2A\right)}{32.2}$$

$$= .00034WRN^2\left(\cos A + \frac{r}{L}\cos 2A\right).$$

PISTON VELOCITY

For the piston velocity V_p at any angle of the crank-pin we have the formula:

$$V_p = \frac{\pi LN}{60 \times 12}\left(\sin A + \tfrac{1}{2}\frac{r}{L}\sin 2A\right).$$

In the following table the value of $\cos A + \frac{r}{L}\cos 2A$, has been calculated for connecting-rod lengths from 3½ to 5½ times crank-radius length.

For instance take $\frac{L}{r}$ to be 4 $\left(\text{then } \frac{r}{L} = \frac{1}{4}\right)$ and take the angle A at 50° forward (which is the same as 310° on the return stroke, *i.e.* 360 − 50 = 310°).

The cosine of 50° = .6428; the cosine of 2 × 50°, *i.e.*, of 100 = .1736 (the cos of angles between 90° and 270° is a negative quantity), we have therefore

$$\text{Cos } 50 - \frac{r}{L}\cos 100° = .6428 - \tfrac{1}{4} \times .1736 = .599$$

If we find this value for angle of 80°, we have

$$\text{Cos } 80° - \frac{r}{L}\cos 160° = .1736 - \tfrac{1}{4} \times .9397 = -\ .061,$$

thus a negative quantity. By looking at the table it is seen that between 70° and 80° the value changes from positive to negative; the reason is that the piston reaches its maximum velocity at a crank angle between 70° and 80° (exact value depending on length of connecting-rod; if rod were infinitely long the piston would reach its maximum speed at 90°) after which the piston begins to slow down, and instead of having to expand energy to set it into motion as is necessary in the beginning of its stroke, it now has a momentum or a force which it exerts upon the crank-pin, thus helping the turning effort of crankshaft until the end of the stroke. At the outer dead center (at 180° of the crank angle) the piston comes to a stop and energy must be imparted to it to set it again into motion. It will again reach its maximum speed between 70° and 80° from the inner dead center, *i.e.*, between 100° and 110° on its return stroke or 180° + 100, or 180° + 110, equals 280° to 290° from the beginning, as shown on the table. Thus at the beginning of the return stroke, the value is again positive, consequently from 180° to 360° the formula becomes $\text{Cos } A - \frac{r}{L}\cos 2A$, hence the plus and minus signs in the table must be reversed on the return stroke. Therefore we wrote the formula in the table $\cos A \pm \frac{r}{L}\cos 2A$, and we term the values found the INERTIA FACTORS.

TABLE IV.—INERTIA FACTORS

Degrees forward	$\left(\cos A \pm \frac{r}{L}\cos 2A\right)$	Conn.-rod length divided by crank-radius $\left(\frac{L}{r}\right)$								Degrees return
		3½	3¾	4	4¼	4½	4¾	5	5½	
0	$1.000+\frac{r}{l}1.000$	1.286	1.267	1.250	1.235	1.222	1.211	1.200	1.183	360
10	$0.9848+\frac{r}{l}0.9397$	1.254	1.236	1.219	1.206	1.194	1.183	1.173	1.156	350
20	$0.9397+\frac{r}{l}0.7660$	1.158	1.144	1.131	1.120	1.110	1.101	1.093	1.079	340
30	$0.866+\frac{r}{l}0.5000$	1.009	0.999	0.991	0.984	0.977	0.971	0.966	0.957	330
40	$0.766+\frac{r}{l}0.1736$	0.816	0.812	0.809	0.807	0.804	0.803	0.801	0.797	320
50	$0.6428+\frac{r}{l}0.1736$	0.593	0.596	0.599	0.602	0.604	0.606	0.608	0.611	310
60	$0.5000-\frac{r}{l}0.5000$	0.357	0.367	0.375	0.382	0.389	0.395	0.400	0.409	300
70	$0.342-\frac{r}{l}0.7660$	0.124	0.138	0.150	0.162	0.172	0.181	0.189	0.203	290
80	$0.1736-\frac{r}{l}0.9397$	−0.095	−0.077	−0.061	−0.047	−0.035	−0.024	−0.014	0.003	280
90	$0.000-\frac{r}{l}1.000$	−0.286	−0.266	−0.250	−0.235	−0.222	−0.210	−0.200	−0.182	270
100	$-0.1736-\frac{r}{l}0.9397$	−0.422	−0.425	−0.408	−0.395	−0.382	−0.372	−0.361	−0.344	260
110	$-0.342-\frac{r}{l}0.7660$	−0.560	−0.546	−0.534	−0.522	−0.512	−0.503	−0.495	−0.481	250
120	$-0.500-\frac{r}{l}0.5000$	−0.643	−0.633	−0.625	−0.618	−0.611	−0.605	−0.600	−0.591	240
130	$-0.6428-\frac{r}{l}0.1736$	−0.693	−0.689	−0.686	−0.684	−0.681	−0.680	−0.677	−0.674	230
140	$-0.7660+\frac{r}{l}0.1736$	−0.716	−0.720	−0.723	−0.725	−0.727	−0.729	−0.731	−0.735	220
150	$-0.8660+\frac{r}{l}0.5000$	−0.723	−0.733	−0.741	−0.748	−0.755	−0.761	−0.766	−0.775	210
160	$-0.9397-\frac{r}{l}0.7660$	−0.722	−0.736	−0.748	−0.760	−0.770	−0.779	−0.786	−0.801	200
170	$-0.9848+\frac{r}{l}0.9397$	−0.717	−0.734	−0.750	−0.764	−0.776	−0.787	−0.797	−0.814	190
180	$-1.000+\frac{r}{l}1.000$	−0.714	−0.734	−0.750	−0.765	−0.778	−0.790	−0.800	−0.818	180

With the aid of this table it is possible to calculate the inertia forces of the reciprocating parts without trigonometrical tables, or a knowledge of trigonometry. As an example take the Class B Quartermaster Truck Motor. In this motor the reciprocating parts, that is to say, the piston, piston pin, and the upper half of the connecting-rod weigh 10 lb. (Piston 6¼ lb., upper end of rod and wrist pin 3¾ lb.)

At 1500 r.p.m. (which is the speed of maximum power for which the engine was designed to run) the piston speed equals 1500 feet per minute as the stroke is 6 inches. The connecting-rod length 13¼ inches and the crank-radius is 3 inches.

$$\text{Therefore } \frac{L}{r} = \frac{13\frac{1}{4}}{3} = 4.42 \text{ or approximately } 4.5.$$

The formula for inertia force *IF*, as was shown before, is

$$\begin{aligned} IF &= .00034WRN^2 \left(\cos A + \frac{r}{L}\cos 2A\right) \\ &= .00034 \times 10 \times .25 \times 1500^2 \left(\cos A + \frac{r}{L}\cos 2A\right) \\ &= 1912.5 \left(\cos A + \frac{r}{L}\cos 2A\right). \end{aligned}$$

For instance at 50°, in the table, under ratio of 4½ we have .604, therefore $IF = 1912.5 \times .604 = 1155$ lb. approximately. At 750 r.p.m. the $IF = .00034 \times 10 \times .25 \times 750^2 \left(\cos 40 + \frac{r}{L}\cos 2A\right) = 478.12 \times .604 = 288$ lb. In this manner the following table was calculated, except that a minus (−) sign was placed in front of these figures since the inertia forces act in opposition to the piston travel at the beginning of the stroke, and *vice versa* after the piston has reached its maximum speed and is decelerating. The figures in ( ), after the numbers, give the inertia force in pounds per square inch of piston area, *i.e.*, the numbers are divided by 17.72 which is the piston area in this engine.

Having found the inertia forces at various crank angles, suppose we now wish to find for the same crank angles, the pressure in the cylinder due to the expansion of the gas following the explosion.

In the Class B Military Truck engine the combustion chamber volume is about 25 per cent. or one-fourth of the total cylinder

TABLE V.—INERTIA FORCES OF RECIPROCATING PARTS OF CLASS B, U. S. TRUCK MOTOR

Degrees forward	At 1500 revolutions per minute		Degrees return
0°	−2340 (−132.0)	+2340 (+132.0)	360
10	−2285 (−128.0)	+2285 (+128.0)	350
20	−2122 (−120.0)	+2122 (+120.0)	340
30	−1868 (−105.0)	+1868 (+105.0)	330
40	−1535 (− 86.5)	+1535 (+ 86.5)	320
50	−1155 (− 65.0)	+1152 (+ 65.0)	310
60	− 744 (− 42.0)	+ 744 (+ 42.0)	300
70	− 329 (− 18.2)	+ 329 (+ 18.2)	290
80	+ 67 (+ 3.78)	− 67 (− 3.78)	280
90	+ 425 (+ 24.0)	− 425 (− 24.0)	270
100	+ 730 (+ 41.1)	− 730 (− 41.0)	260
110	+ 980 (+ 55.2)	− 980 (− 55.2)	250
120	+1168 (+ 66.0)	−1168 (− 66.0)	240
130	+1300 (+ 73.4)	−1300 (− 73.4)	230
140	+1389 (+ 78.4)	−1389 (− 78.4)	220
150	+1442 (+ 81.5)	−1442 (− 81.5)	210
160	+1472 (+ 83.0)	−1472 (− 83.0)	200
170	+1483 (+ 83.6)	−1483 (− 83.6)	190
180	+1485 (+ 83.8)	−1485 (− 83.8)	180

TABLE V.—INERTIA FORCES OF RECIPROCATING PARTS OF CLASS B, U. S. TRUCK MOTOR

Degrees forward	At 750 revolutions per minute		Degrees return
0°	−585	+585	360°
10	−571	+571	350
20	−531	+531	340
30	−466	+466	330
40	−384	+384	320
50	−288	+288	310
60	−186	+186	300
70	− 82	+ 82	290
80	+ 17	− 17	280
90	+106	−106	270
100	+182	−182	260
110	+244	−244	250
120	+292	−292	240
130	+325	−325	230
140	+347	−347	220
150	+360	−360	210
160	+368	−368	200
170	+370	−370	190
180	+371	−371	180

volume, which is the same as a ratio of compression of $\frac{4}{1} = 4$. (See Chapter III, where it was shown that the ratio of compression is $\frac{V_1}{V_2}$).

With such a ratio of compression and an initial pressure of 13 lb. (the average), we expect to obtain a compression pressure of about 65 lb. gauge or about 80 lb. absolute.

Let us assume that the pressure at the beginning of the expansion stroke is 3½ times that of compression or 280 lb. and that the expansion curve follows the equation

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{1.3}; \text{ In this case } P_1 = 280$$

V_1 = the volume of the clearance, and V_2 = the volume of clearance + the increase of volume due to the piston positions. Instead of taking the entire volume we may take the volume as unity (1 cubic inch for instance) for each inch length of piston stroke. The stroke is 6 inches and we thus take the volume as 6; the clearance V_2 being 25 per cent. of the total volume V_1, we have $V_1 - V_2 = 6$, and V_1 must clearly be 8, and $V_2 = 2$, in order that V_2 = 25 percent. of V_1. Suppose we wish to find P_2 for piston positions corresponding to the same crank positions as given in the last table for the Inertia Forces. (It would of of course be simpler to find the pressure for certain equi-distant spaces of piston travel.)

The following table gives the positions of the piston for an entire revolution of the crank and for a ratio of connecting-rod length to crank length from 4 to 5.

The quantities in this table are expressed in fractions of the total stroke. To find the piston travel for an angle of 50° on the out stroke for example, when the ratio of connecting-rod to crank length equals 4.5; the table gives .2115 and since the total piston stroke in this motor is 6 inches, the piston will have moved $.2115 \times 6 = 1.269$ inches.

To find P_2 per square inch when the piston has traveled this distance, we have $P_2 = P_1\left(\frac{V_1}{V_2}\right)^{1.3}$

$V_1 = 2$, as we have seen before.

$V_2 = 2 + 1.269 = 3.269$,

Therefore $P_2 = 280 \left(\frac{2}{3.269}\right)^{1.3} = 280\ (.611)^{1.3} = 147.2$ pounds per square inch.

TABLE VI.—CRANK ANGLES AND CORRESPONDING PISTON POSITIONS[1]

'Out' denotes the stroke toward the crankshaft.

'In' denotes the stroke away from the crankshaft.

Crank angles in degrees	Connecting-rod to crank ratio = 4		Connecting-rod to crank ratio = 4.5		Connecting-rod to crank ratio = 5	
	Out	In	Out	In	Out	In
2	0.0004	0.0002	0.0004	0.0003	0.0004	0.0003
4	0.0015	0.0010	0.0015	0.0010	0.0015	0.0010
6	0.0034	0.0021	0.0033	0.0021	0.0033	0.0022
8	0.0061	0.0036	0.0060	0.0038	0.0058	0.0039
10	0.0095	0.0057	0.0092	0.0059	0.0091	0.0061
12	0.0136	0.0082	0.0133	0.0087	0.0131	0.0088
14	0.0185	0.0112	0.0181	0.0116	0.0178	0.0119
16	0.0241	0.0146	0.0236	0.0152	0.0232	0.0156
18	0.0304	0.0185	0.0298	0.0192	0.0293	0.0197
20	0.0375	0.0228	0.0367	0.0237	0.0360	0.0243
22	0.0452	0.0276	0.0442	0.0286	0.0434	0.0294
24	0.0536	0.0329	0.0524	0.0340	0.0515	0.0349
26	0.0627	0.0385	0.0613	0.0399	0.0602	0.0410
28	0.0724	0.0447	0.0708	0.0462	0.0696	0.0475
30	0.0827	0.0513	0.0809	0.0531	0.0796	0.0545
32	0.0936	0.0583	0.0916	0.0603	0.0901	0.0619
34	0.1051	0.0658	0.1029	0.0680	0.1012	0.0698
36	0.1172	0.0738	0.1147	0.0762	0.1128	0.0782
38	0.1298	0.0822	0.1271	0.0848	0.1250	0.0870
40	0.1430	0.0910	0.1400	0.0939	0.1377	0.0962
42	0.1565	0.1002	0.1534	0.1034	0.1509	0.1059
44	0.1707	0.1099	0.1673	0.1134	0.1646	0.1161
46	0.1853	0.1201	0.1816	0.1237	0.1787	0.1267
48	0.2003	0.1306	0.1963	0.1345	0.1392	0.1377
50	0.2156	0.1416	0.2115	0.1458	0.2081	0.1491
52	0.2314	0.1530	0.2270	0.1574	0.2234	0.1609
54	0.2474	0.1648	0.2428	0.1695	0.2391	0.1732
56	0.2638	0.1769	0.2589	0.1819	0.2550	0.1858
58	0.2805	0.1896	0.2754	0.1947	0.2713	0.1988
60	0.2974	0.2026	0.2920	0.2079	0.2878	0.2122
62	0.3146	0.2159	0.3090	0.2215	0.3046	0.2260
64	0.3320	0.2297	0.3262	0.2355	0.3215	0.2401

[1] From The Automobile Trade Directory.

TABLE VI.—CRANK ANGLES AND CORRESPONDING PISTON POSITIONS (*Continued*)

'Out' denotes the stroke toward the crankshaft.
'In' denotes the stroke away from the crankshaft.

Crank angles in degrees	Connecting-rod to crank ratio = 4		Connecting-rod to crank ratio = 4.5		Connecting-rod to crank ratio = 5	
	Out	In	Out	In	Out	In
66	0.3495	0.2438	0.3435	0.2498	0.3387	0.2545
68	0.3672	0.2582	0.3610	0.2644	0.3561	0.2693
70	0.3850	0.2730	0.3786	0.2794	0.3735	0.2844
72	0.4028	0.2881	0.3963	0.2947	0.3911	0.2998
74	0.4208	0.3036	0.4141	0.3103	0.4088	0.3155
76	0.4388	0.3193	0.4319	0.3261	0.4266	0.3315
78	0.4568	0.3353	0.4498	0.3422	0.4444	0.3477
80	0.4747	0.3516	0.4677	0.3586	0.4622	0.3642
82	0.4927	0.3682	0.4857	0.3753	0.4799	0.3809
84	0.5105	0.3849	0.5034	0.3921	0.4977	0.3978
86	0.5283	0.4019	0.5211	0.4091	0.5154	0.4149
88	0.5460	0.4191	0.5388	0.4263	0.5330	0.4321
90	0.5635	0.4365	0.5563	0.4437	0.5505	0.4495
92	0.5802	0.4540	0.5737	0.4612	0.5670	0.4670
94	0.5981	0.4787	0.5909	0.4789	0.5851	0.4846
96	0.6151	0.4895	0.6079	0.4966	0.6022	0.5028
98	0.6318	0.5073	0.6247	0.5143	0.6191	0.5201
100	0.6484	0.5253	0.6414	0.5323	0.6358	0.5378
102	0.6647	0.5432	0.6578	0.5502	0.6523	0.5556
104	0.6807	0.5612	0.6739	0.5681	0.6685	0.5734
106	0.6964	0.5792	0.6897	0.5859	0.6845	0.5912
108	0.7119	0.5972	0.7053	0.6037	0.7002	0.6082
110	0.7270	0.6150	0.7206	0.6214	0.7156	0.6265
112	0.7418	0.6328	0.7356	0.6390	0.7307	0.6439
114	0.7562	0.6505	0.7502	0.6565	0.7455	0.6613
116	0.7703	0.6680	0.7645	0.6738	0.7599	0.6785
118	0.7841	0.6854	0.7785	0.6910	0.7740	0.6954
120	0.7974	0.7026	0.7921	0.7080	0.7878	0.7122
122	0.8104	0.7195	0.8053	0.7246	0.8012	0.7287
124	0.8230	0.7362	0.8181	0.7411	0.8142	0.7450

Table VI.—Crank Angles and Corresponding Piston Positions (*Concluded*)

'Out' denotes the stroke toward the crankshaft.
'In' denotes the stroke away from the crankshaft.

Crank angles in degrees	Connecting-rod to crank ratio = 4		Connecting-rod to crank ratio = 4.5		Connecting-rod to crank ratio = 5	
	Out	In	Out	In	Out	In
126	0.8352	0.7526	0.8305	0.7572	0.8268	0.7609
128	0.8470	0.7686	0.8426	0.7730	0.8391	0.7766
130	0.8584	0.7844	0.8542	0.7885	0.8509	0.7919
132	0.8694	0.7997	0.8655	0.8037	0.8623	0.8068
134	0.8799	0.8147	0.8763	0.8184	0.8733	0.8213
136	0.8901	0.8293	0.8866	0.8327	0.8839	0.8354
138	0.8998	0.8434	0.8966	9.8466	0.8941	0.8491
140	0.9090	0.8570	0.9061	0.8600	0.9038	0.8623
142	0.9178	0.8702	0.9152	0.8729	0.9130	0.8750
144	0.9262	0.8828	0.9238	0.8853	0.9218	0.8872
146	0.9342	0.8949	0.9320	0.8971	0.9302	0.8988
148	0.9417	0.9064	0.9397	0.9084	0.9381	0.9099
150	0.9487	0.9173	0.9469	0.9191	0.9455	0.9204
152	0.9553	0.9276	0.9538	0.9292	0.9525	0.9304
154	0.9617	0.9373	0.9601	0.9387	0.9590	0.9398
156	0.9671	0.9464	0.9660	0.9476	0.9651	0.9485
158	0.9724	0.9548	0.9714	0.9558	0.9706	0.9566
160	0.9772	0.9625	0.9763	0.9633	0.9757	0.9640
162	0.9815	0.9696	0.9806	0.9702	0.9803	0.9707
164	0.9854	0.9759	0.9848	0.9764	0.9844	0.9767
166	0.9888	0.9815	0.9884	0.9819	0.9881	0.9822
168	0.9918	0.9864	0.9913	0.9867	0.9912	0.9869
170	0.9943	0.9905	0.9941	0.9908	0.9939	0.9909
172	0.9963	0.9939	0.9962	0.9940	0.9961	0.9942
174	0.9979	0.9966	0.9979	0.9967	0.9978	0.9967
176	0.9990	0.9985	0.9990	0.9985	0.9990	0.9985
178	0.9998	0.9936	0.9997	0.9996	0.9097	0.9996
180	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000

(This equation is solved by logarithms as explained in Chapter II.) To find the total pressure on the piston head we multiply the pressure per square inch by the area of the piston. (The piston of this motor has a diameter of 4.75 inches and therefore

an area of 17.72 square inches.) The pressure in this case would be 147.2 × area = 147.2 × .7854 × 4.75^2 = 2630 lbs.

(All the figures in the following table were calculated with the slide rule.)

TABLE VII.—GAS PRESSURE ON THE PISTON HEAD OF THE CLASS B MILITARY TRUCK ENGINE. (PRESSURE AT BEGINNING OF EXPANSION STROKE ASSUMED TO BE 280 LBS. PER SQUARE INCH)

Crank angle degrees	$P_2 = 280\left(\frac{V_1}{V_2}\right)^{1.3}$	P_2 per sq. in.	Total pressure on piston head = P_2 × piston area
0	$280\left(\frac{2}{2}\right)^{1.3}$	280	4965
10	$280\left(\frac{2}{2.055}\right)^{1.3}$	270.95	4800
20	$280\left(\frac{2}{2.220}\right)^{1.3}$	244.13	4330
30	$280\left(\frac{2}{2.534}\right)^{1.3}$	206.08	3650
40	$280\left(\frac{2}{2.84}\right)^{1.3}$	177.74	3240
50	$280\left(\frac{2}{3.269}\right)^{1.3}$	147.2	2630
60	$280\left(\frac{2}{3.756}\right)^{1.3}$	123.00	2180
70	$280\left(\frac{2}{4.271}\right)^{1.3}$	104.00	1845
80	$280\left(\frac{2}{4.806}\right)^{1.3}$	89.9	1590
90	$280\left(\frac{2}{5.337}\right)^{1.3}$	78.00	1380
100	$280\left(\frac{2}{5.848}\right)^{1.3}$	69.8	1275
110	$280\left(\frac{2}{6.323}\right)^{1.3}$	62.5	1115
120	$280\left(\frac{2}{6.752}\right)^{1.3}$	57.5	1018
130	$280\left(\frac{2}{7.125}\right)^{1.3}$	53.6	950
140	$280\left(\frac{2}{7.436}\right)^{1.3}$	51.00	905
150	$280\left(\frac{2}{7.681}\right)^{1.3}$	48.6	862
160	$280\left(\frac{2}{7.957}\right)^{1.3}$	46.9	831
170	$280\left(\frac{2}{7.964}\right)^{1.3}$	46.4	823
180	$280\left(\frac{2}{8}\right)^{1.3}$	46.1	818

RESULTANT PRESSURES IN CLASS B ENGINE

Knowing the inertia forces in pounds per square inch of piston area and the pressure due to the expanding gas, we can draw curves showing the resultant force on the piston during the four strokes, which with the centrifugal force will enable us to judge of the bearing pressures on the wrist pin, the crank-pin and the crankshaft.

Figure 93 shows curves plotted during the four strokes taking into account the forces due to compression, expansion, and the

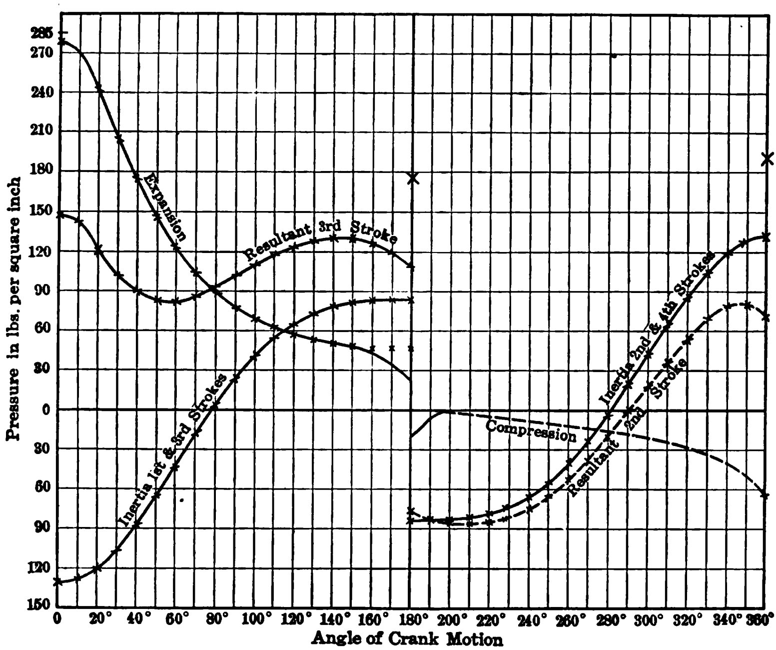


FIG. 93.—Curves showing the forces due to compression and expansion of the gas and inertia of the reciprocating parts of the Class B engine at 1500 revolutions per minute.

inertia of the reciprocating parts (the pressure due to the centrifugal force has not been added). The suction and the exhaust curves are not shown as they are very small. It should be noted that all the forces acting in opposition to the piston travel, or the forces which tend to retard the crankshaft have been drawn below the zero line, while those assisting the crankshaft, thus the positive forces, are drawn above the zero line. On the first

and fourth strokes the resultant pressure curves (disregarding the centrifugal force) are practically the same as the inertia curves. On the second stroke the compression pressure must be deducted from the inertia forces since the latter act in opposition to the compression toward the end of the compression stroke. On the third stroke at first the inertia acts in opposition to the gas pressure until approximately 78° of the crank motion. Thereafter the inertia forces are added to the pressure for the rest of this stroke. The inertia and expansion curves were drawn from the values found in the preceding table. The compression pressure follows the law (see Chapter III) PV^n = constant, and it is a negative force trying to retard the piston.

The centrifugal force is a "positive" force and always acts outward from the crankshaft and therefore at the ends of the strokes it is simply added to the resultant pressure. At 1500 revolutions per minute it was shown that the centrifugal force of the Class B engine was 1052 lb. which divided by the area of the piston, *i.e.*, $\frac{1052}{17.72} = 59.5$ lb. Adding 59.5 to the resultant pressure at the end of the expansion stroke, will give us a pressure of 176 lb. and adding it to the resultant at the end of the fourth stroke, gives a pressure of 190 lb. per square inch of piston area, or a total of $190 \times 17.72 = 3367$ lb.

It is thus seen that the inertia and the centrifugal forces overbalance the explosion and compression forces and the bearing area must be ample for these maximum pressures, which in this case as stated, amount to 3367 lb. The centrifugal force always acts radially outward, and the inertia forces also act outward at the end of the stroke, therefore the crank-pins always show the most wear on the inner side in high-speed engines. For other positions of the crank, than at the dead center, the inertia force, and the gas pressure, does of course, not act radially outward and would therefore result in a decrease of the bearing pressure.

In the class B engine the crank-pins are $2\frac{3}{8}$ inches in diameter and 3 inches long. Deducting the fillets of $\frac{1}{8}$ inch on each side, leaves a bearing length of $2\frac{3}{4}$ inches, and a projected area of $2\frac{3}{4} \times 2\frac{3}{8} = 6.53$ square inches. The maximum pressure on the pins is therefore $\frac{3367}{6.53} = 515$ lb., which is amply sufficient for an engine designed for hard service. In high-speed automobile engines, the maximum crank-pin bearing pressure is frequently as high as 900 to 1000 lb. per square inch.

CHAPTER XIII

FLYWHEELS

In all prime movers relying on pressure in a cylinder acting on pistons, or where reciprocating motion is transformed into revolving motion, intermittent impulses of varying strength are applied to the crankshaft which therefore has a tendency to fluctuate in speed. In four-cycle engines, where there is only one power stroke in each four strokes of the piston, this tendency is still greater.

To keep this speed fluctuation within certain limits, a flywheel of sufficient weight is necessary. Its office is to carry the crankshaft around during the three idle strokes of the piston, *i.e.*, during the suction, compression and exhaust strokes; thus it must store a certain amount of energy of the engine during the power stroke and restore it during the three idle strokes.

It is also found in practice, that a heavier flywheel permits a more rapid acceleration of the car, and it also facilitates the cranking of the motor.

KINETIC ENERGY OF THE FLYWHEEL

Every body or mass in motion contains a certain amount of energy; if it were not for gravity and friction it could return the same amount of energy which was necessary to set it into motion. This energy is called the Kinetic energy.

The kinetic Energy K stored in the flywheel $= \frac{WV^2}{2g}$ ft. lb. per second: W being the weight in lbs., V the speed in feet per second, and g the acceleration due to gravity which is 32.16 feet per second per second (usually 32.2 is used).

If r is the distance of the center of the flywheel rim from the center of the crankshaft, in inches, then $\frac{r}{12}$ will be the radius in feet, and the angular speed will be $\frac{N}{60}$ revolutions per second, where N is the number of revolutions per minute. The lineal

velocity of the center of the flywheel rim $V = \frac{2\pi rN}{12 \times 60}$, and the energy stored up in ft. lb. is

$$K = \frac{WV^2}{2g} = \frac{W\,4\pi^2r^2N^2}{2g \times 12^2 \times 60^2} = \frac{Wr^2N^2}{26{,}262g}\text{ ft. lb.} = \frac{Wr^2N^2}{854{,}550}.$$

FLYWHEEL WEIGHT

Ordinary cast iron, of which flywheels are usually made, has a specific gravity of 7.08. The weight of 1 cubic foot averages 442 lb., and therefore 1 cubic inch weighs $\frac{442}{12 \times 12 \times 12} = .255$ lb.

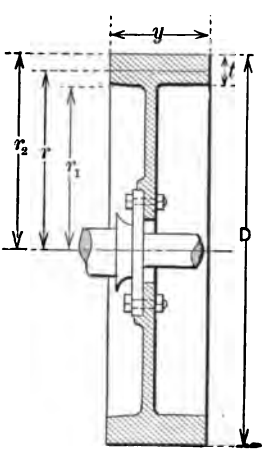


FIG. 94.—Flywheel.

If t is the thickness of the flywheel flange or rim (Fig. 94), and y the flywheel face width on the periphery, in inches, then, for practical purposes the number of cubic inches in the flywheel $= 2\pi rty$; r is the mean radius of the flywheel rim and may be determined as follows: $r = \frac{r_1 + r_2}{2}$, where $r_1 =$ the radius of the internal diameter of the flange, $r_2 =$ the radius of the outside diameter. The weight of the rim, $W = 2\pi rty \times .255 = 1.6\, rty$ lb. Substituting this value for W in the formula for the energy stored, we have:

$$K = \frac{1.6tyr^3N^2}{26{,}262g} = \frac{tyr^3N^2}{16{,}413g}\text{ft. lb.}$$

The mean radius of the flywheel rim (the radius of gyration) cannot always be so easily determined. For instance in Fig. 35*a* may be seen the flywheel of the Haynes Light Six Motor. In such a section it is difficult to find the center of the rim, *i.e.*, the center of gravity of the rim.

Figure 95 shows a similar irregular-shaped flange or rim.

The kinetic energy in such a flywheel is determined as follows: First lay out the section A and find its center of gravity, which is on line a, with radius of gyration r; next lay out section B and mark its center line of gravity b which is the radius of gyration r_1 of this section of the flywheel; then lay out section C with its

center of gravity c and its radius of gyration r_2. We then have the kinetic energy of

$$A = \frac{tyr^3N^2}{16{,}413g}\text{ft. lb., and of}$$

$$B = \frac{t_1y_1r_1^3N^2}{16{,}413g};\text{ where } t_1 \text{ and } y_1 \text{ are the}$$

thickness and the width respectively of section B, and the kinetic energy of $C = \dfrac{t_2y_2r_2^3N^2}{16{,}413g}$, where t_2 and y_2 are the thickness and the width of section C.

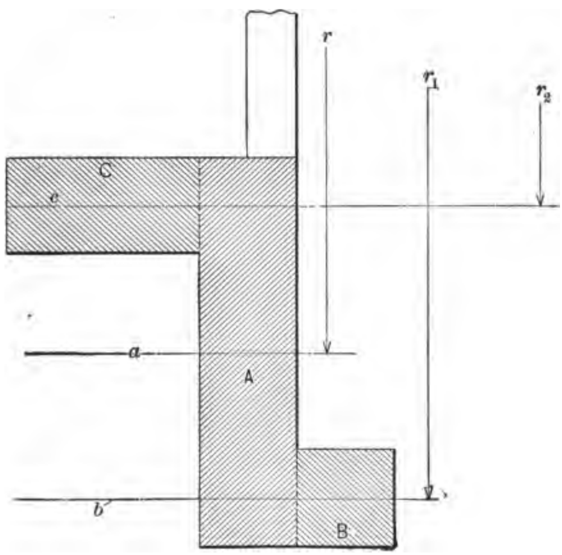


Fig. 95.

The kinetic energy K of a flywheel of this section is therefore

$$K = \frac{tyr^3N^2}{16{,}413g} + \frac{t_1y_1r_1^3N^2}{16{,}413g} + \frac{t_2y_2r_2^3N^2}{16{,}413g}$$
$$= \frac{(tyr^3 + t_1y_1r_1^3 + t_2y_2r_2^3)N^2}{16{,}413g}$$

that is to say, the kinetic energy of each section has to be found separately and added together will give the total kinetic energy in the flywheel.

In figuring the kinetic energy as a rule only the flywheel rim

is calculated, not the spokes or the web, but they too, have a small flywheel effect.

The flywheel weight must be sufficient to keep the speed fluctuations within a certain limit. To do this it must store more power than merely necessary to carry the motor around during the idle strokes, since it also must supply power to overcome the compression pressure, the suction and the exhaust back-pressure. The following table gives the approximate weights which may be used for the flywheel rim, with various radii of rim.

r = the radius of gyration in inches.

Bore	Weight of flywheel rim in pounds				
	r = 5"	r = 6"	r = 7"	r = 8"	r = 9"
2½	25	20	15		
3	60	40	30		
3½	100	70	50	40	
4	150	105	80	62	48
4½	....	150	110	82	65
5	....	....	145	110	88

These weights are suitable for engines having a ratio of stroke to bore as given in Chapter V. When the stroke is longer this weight is slightly increased and vice versa.

FLYWHEEL DIAMETER

The outside diameter of the flywheel varies from about 11 inches for very small motors whose stroke is 3½ inches, to about 19½ inches for engines of 6 inch stroke. On the average the proportion of the outside diameter D is from 3.2 to 3.3 times the length of stroke.

The following standard sizes for flywheel housings were adapted by the Society of Automotive Engineers, and were published in the S. A. E. data sheet.

TABLE VIII.—DISK-CLUTCH HOUSINGS (SEE FIG. 96)

Size No.	A	B	C
1	20⅛	21¾	20⅞
2	17⅝	19¼	18⅜
3	16⅛	17¾	16⅞
4	14¼	15⅞	15
5	12⅜	14	13⅛

All the dimensions are in inches. The dimensions shown on the drawing indicate clearances for disc clutch and do not apply to size No. 5.

TABLE IX.—CONE-CLUTCH HOUSINGS (SEE FIG. 97)

Size No.	A	B	C	E
2	17⅝	19¼	18⅜	15¾
3	16⅛	17¾	16⅞	13¾

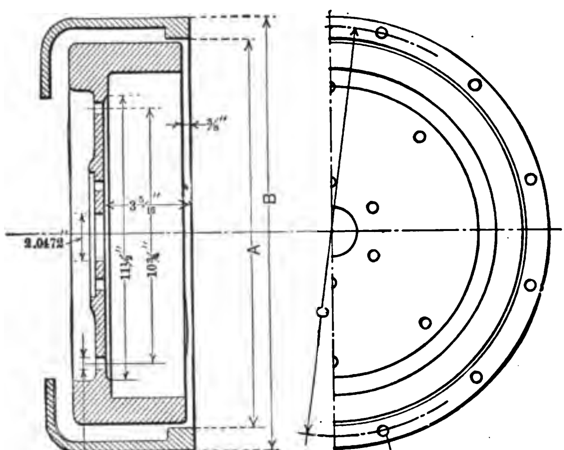


FIG. 96.—Flywheel and disk-clutch housing.

All the dimensions are in inches. The dimensions shown on the drawing indicate clearances for cone clutches and tail shaft details. These housings will, of course, limit the diameter of the flywheel.

In working out an example let us assume it is required to find the weight of a flywheel rim for a six-cylinder engine having a bore of 4½ inches, stroke of 6 inches, and a mean diameter of flywheel rim of 16 inches. Therefore the mean radius of gyration, that is to say the distance from the center of the flywheel to the center of the flywheel rim, is 8 inches.

Looking at our table we find the weight as 82 lb. but as the ratio of stroke to bore is somewhat greater than 1.25 to 1 (see page 34) we increase this weight somewhat and call it in round numbers 90 lb.

When proportioning the area of a flywheel rim, it is recommended to make the face rather wide and the rim not very deep radially; in this manner, the mean radius (the radius of gyration), will approach the radius of the outside of the rim.

In Chapter XIV it is shown that the centrifugal force of a rotating mass is $CF = .00034WRN^2$; where N is the number of revolutions per minute, and R the radius of rotation in feet. If R = the radius, in inches, we have

$$CF = \frac{.00034WrN^2}{12}.$$

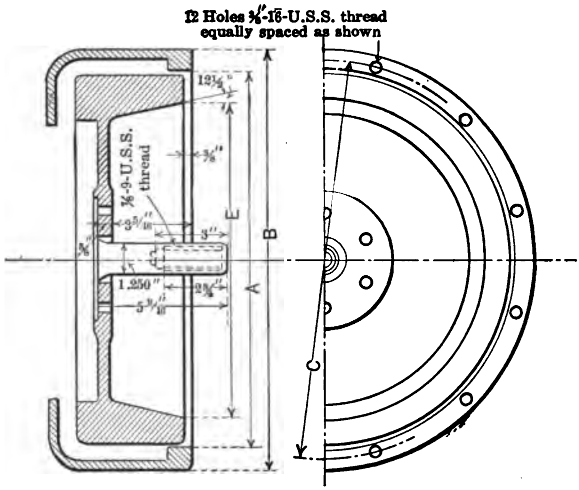


Fig. 97.—Flywheel and cone-clutch housing.

The weight of the flywheel rim is the number of cubic inches multiplied by the weight per cubic inch, and this as shown before equals $2\pi rty \times .255$. By substituting this value for W in the above equation, we have

$$CF = \frac{.00034rN^2}{12} 2\pi rty \times .255 = \frac{\pi r^2N^2ty}{69200}\text{ lb.}$$

The force tending to rupture the flywheel is $= CF\frac{1}{2}\frac{2r}{\pi r}$, *i.e.*, the centrifugal force multiplied by one-half times the ratio of the diameter to one-half the circumference. If F_R is the force

tending to rupture the flywheel, by substituting the value for CF found before, we have:

$$F_R = \frac{\pi r^2 N^2 ty}{69200} \tfrac{1}{2} \frac{2r}{\pi r} = \frac{r^2 N^2 ty}{69200}$$

In order to find the actual stress in the metal per unit of area, we have to divide F_R by two sections of the flywheel at opposite sides of the diameter since two sections will resist rupture, (see Fig. 26, and the description on page 35).

The stress S tending to rupture each square inch of metal is:

$$S = \frac{\frac{r^2 N^2 ty}{69200}}{2ty} = \frac{r^2 N^2 ty}{2ty\ 69200} = \frac{r^2 N^2}{138400} \text{ lb. per sq. inch.}$$

The tensile strength of cast iron varies from about 10,000 to 30,000 lb. per square inch. The material used in flywheels let us say has a tensile strength of 18,000 lb. Knowing this value we may calculate the speed at which the wheel would burst. By first substituting 18,000 for S we have

$$18{,}000 = \frac{r^2 N^2}{138400} \text{lb. per square inch,}$$

and the bursting speed would therefore be, when solving this equation for N

$$N^2 = \frac{18{,}000 \times 138{,}400}{r^2}; \text{ from which } N = \frac{50{,}000}{r}.$$

If the radius is 8 inches as in the last example,

$$N = \frac{50{,}000}{8} = 6250 \text{ r.p.m. approx.}$$

The web used between the hub and the rim of the flywheel will also add to the strength, making this bursting speed even higher.

Automobile engine manufacturers as a rule dispense with keys for fastening the wheel to the crankshaft but provide a flange on the crankshaft, varying from 3/8 inch to 5/8 inch in thickness, which is recessed into the web of the flywheel and bolted thereto. The flywheel web will vary in thickness from 3/8 inch for very small engine, having a bore of 2½ inch, to 5/8 inch for engines having a bore of 5 inches.

When the flywheel is keyed to the crankshaft as in Fig. 85, the shaft is usually tapered as seen here, according to the S. A. E. standard taper of 1½ inch per foot. The key is usually made from ¼ to ⅕ the shaft diameter.

CHAPTER XII

CRANKCASES

The structure which supports the cylinders, the bearings for the crankshaft and the camshaft, which encloses the lower working mechanism of the motor, and supports the latter on the frame, is called the crankcase. It is good practice to allow not less than ¼ inch clearance where the connecting rod head and the case come nearest together during the motion of the former; the smallest clearance is usually at the sides of the case. The top of the crankcase is either cast in one with the cylinders, or the cylinders are bolted to it. Since the separate cylinder head has come into use, in many cases the crankcase entire is cast with the cylinders en block, while in other cases only the upper half of the case is cast integral with the cylinders (see Fig. 30).

When the crankcase is cast en block with the cylinders, it is made of cast iron. On the other hand, when the cylinder castings are bolted to it, it is usually made of aluminum. A few years ago, aluminum alloy was almost universally used for crankcases. One reason for using aluminum was its lightness; on the other hand, by casting the crankcase en block with the cylinders the heavy flanges or joints between cylinder and crankcase are obviated, thus saving this weight, and for this reason, the difference in weight when using an aluminum crankcase, or one of cast iron integrally cast with the cylinders, is usually not very great. In commercial vehicles where a few extra pounds are not so important, it is more economical to use cast iron, especially when the price of aluminum is high.

Ordinarily all the crankshaft bearings are supported from the top half of the case (at *B*, Fig. 98*i*, 99*c*, 99*e*) and the lower half of the case serves merely as a protection for the working parts and contains a supply of oil.

In Fig. 62, which shows an inverted view of the upper half of the White crankcase, *bb* show the crankshaft bearings, *f* the housng for the flywheel, *s* the flange to which the lower half of the crankcase is bolted, *c* the bolts by which the bearing caps are held to the crankshaft bearings.

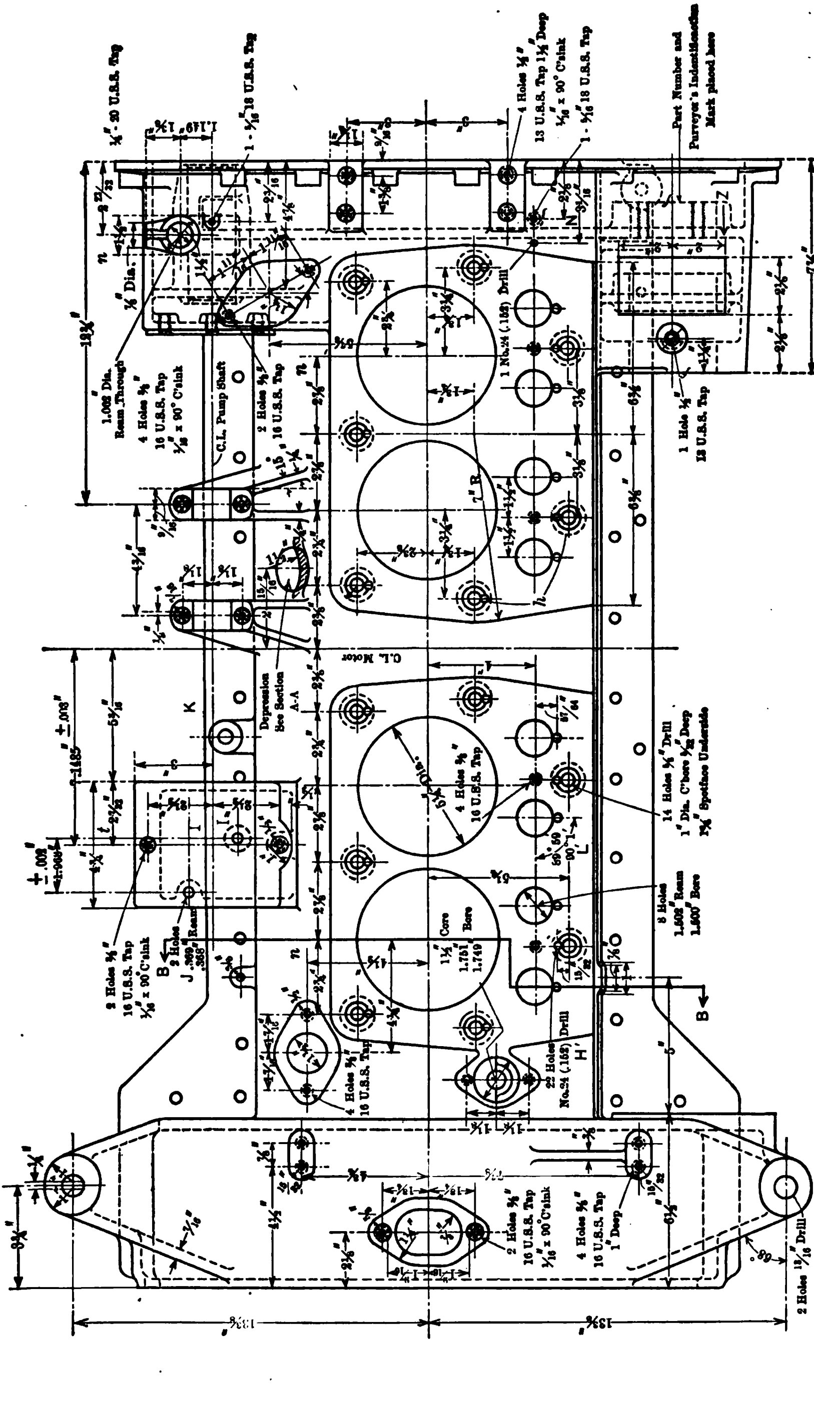


Fig. 98.—Plan view of upper half of crankcase of Class B military truck engine.

At the present time many engines have barrel-shaped crankcases as shown in Fig. 53; an inspection hole, with cover plate in position, being seen at the right hand side, by which the connecting rods and crankshaft bearings can be examined. With this type of crankcase there is sometimes a removable receptacle at the bottom, containing the oil trough. In Fig. 53 (Reo) the oil receptacle *o* is part of the crankcase casting but taps or plugs *p* are provided for draining the oil. In other engines the crankcase is made in two halves as in Fig. 99*d* to 99*g*, the Pierce Arrow (24-valve engine $4\frac{1}{2} \times 5\frac{1}{2}$) aluminum crankcase.

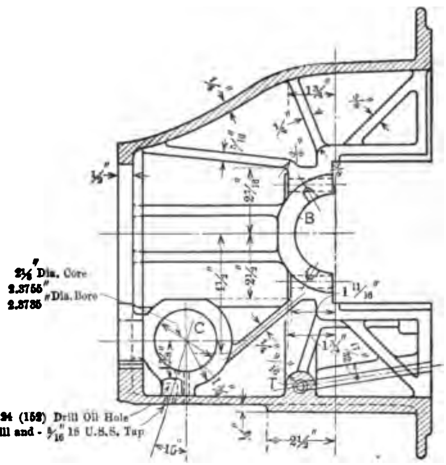


FIG. 98*a*.—Cross-section through *B-B* of Fig. 98.

When the camshaft is placed in a tunnel formed in the casing, the crankcase is enlarged where the tunnel is formed (see Fig. 29, and Figs. 99*d* to 99*g*).

Figures 36 and 36*a* give a good view of the crankcase of the Waukesha Motor. The crankcase is of aluminum made in two parts, bolted together in the center. The thickness of the metal in the upper half is ¼″; in the lower half 3/16″, with an extra thickness at the joint. "*a*" shows the rear supports by which the motor is resting on the chassis, and fastened to it by two ½ bolts in each arm. The front is attached at one point to a cross member of the frame. Note the number of plugs *p* for draining the oil underneath each cylinder. *PP* show the partitions which have to be used in some form or other, when the cylinders are splash lubricated. (See Chapter on "Engine Lubrication.") In

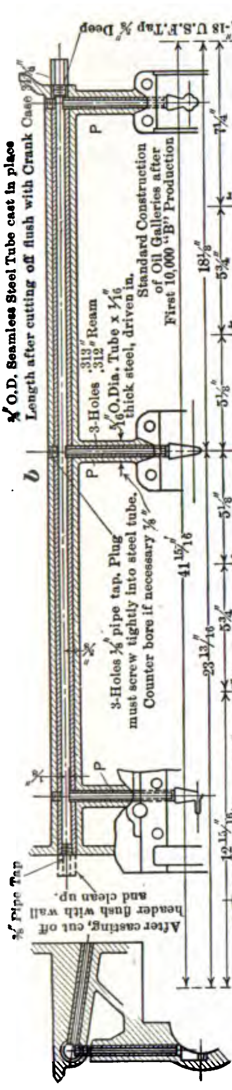


FIG. 98b.—Tubing for the distribution of oil.

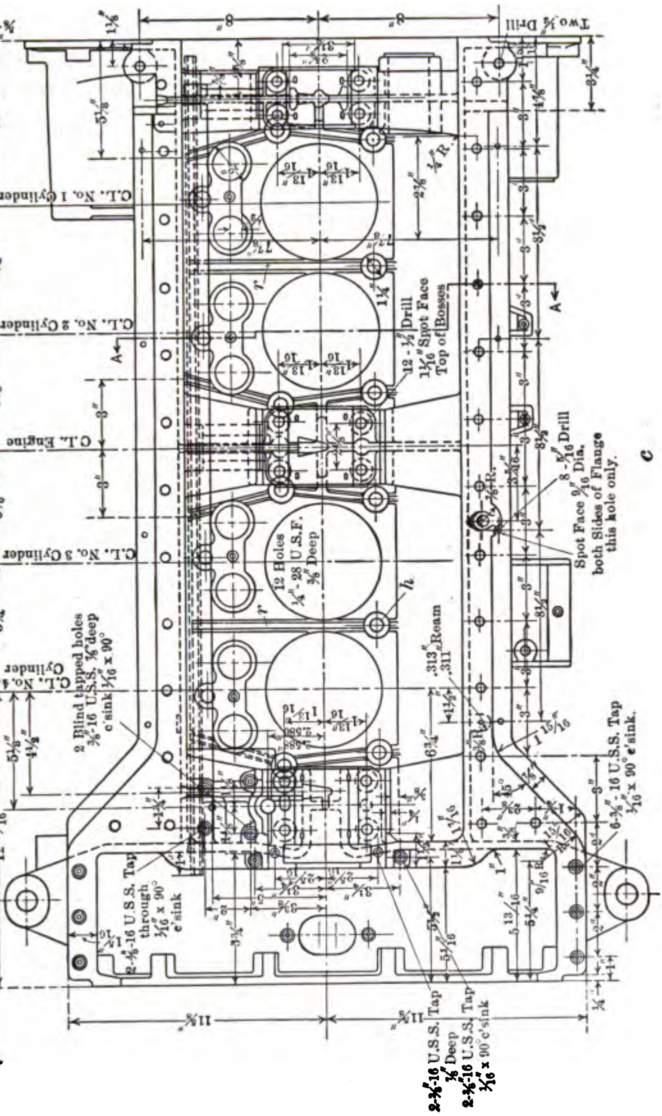


FIG. 98c.—Inverted view of Fig. F.

aluminum cases these partitions are about $3/16$ inches thick. Unless such partitions were provided, when the motor vehicle is moving up an incline all the oil would flow to the rear of the crankcase.

In Figs. 35 and 35*a* the upper half of the crankcase including housing for the flywheel is cast of aluminum, and the lower half of the case is made of sheet steel (steel stamping) as can be seen from the figures. Instead of partitions between the cylinders

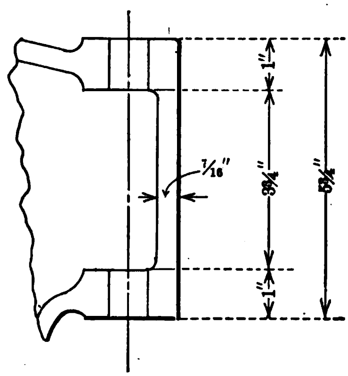


Fig. 98*d*.—Crankcase supporting arm of Class B engine.

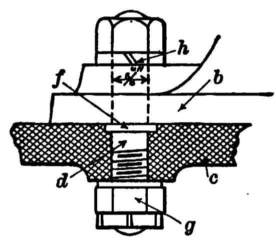
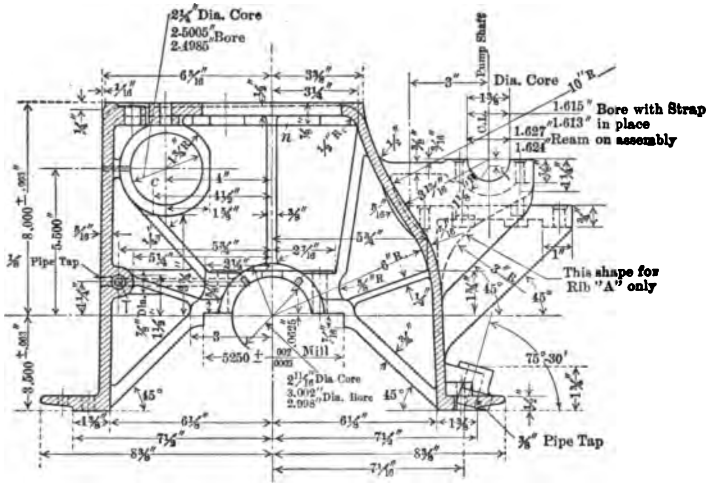


Fig. 98*e*.—Attachment of cylinders to crankcase.

depressions *D* are provided to prevent the oil from flowing to the rear or the front end of the crankcase when the vehicle is traveling up or down an incline.

Figure 98 shows a top view of the upper half of the crankcase of the Class *B* Military Truck Engine with the cylinders removed. Figure 98*a* shows a cross-sectional view through *B-B*. Figure 98*b* shows the steel tubing provided in the crankcase for the distribution of the oil for lubrication. This tube is shown at *T* in Fig. 98. Figure 98*c* shows an inverted view of Fig. 98. It shows the bearing for the crankshaft and the ribs for strengthening the case. There are seven $5/8''$ holes *h* for bolting each pair of cylinders to the case. These holes are spot faced on the underside. The thickness of the aluminum case, where the bolts are, is $7/8''$, which includes the boss (see Fig. 98*e*). "*c*" is the aluminum case, *b* the cylinder casting, and *d* the special bolt which, as is seen, fits into the crankcase from above before the cylinder is placed in position. The shoulder *f* of the bolt is counter-sunk into the aluminum case. There is a castellated

nut g at the bottom (inside the case), and a simple nut with lock washer h on the top, for bolting down the cylinder. Figure 98f is a cross section on Fig. 98c; Fig. 98g is a longitudinal section through the crankcase, and Fig. 98h is a side elevation. These views give the upper half of the crankcase to which the cylinder castings are bolted. Figures 34 and 34a show the lower half of the case. The cylinders must be securely bolted to the crank case (with bolted on cylinders), since the explosion will tend to force them away, on account of the reaction set up. The torque



SECTION A-A *f*

FIG. 98*f*.

reaction, of which we spoke in Chapter XI, tends to pull the cylinders away from the case, and it also strains the supporting arms, by which the motor is fastened to the frame.

In bolting the cylinders to the crankcase, as a rule the bolts are in some way fastened to the crankcase so that it is only necessary to place the nut on the outside of the cylinder flange. Threading the aluminum crankcase flange is not recommended since the thread will easily pull out from the aluminum, therefore the construction shown in Fig. 98e, or a bolt having a head at the bottom is recommended. However, the bolt may be slightly enlarged and threaded at the head end for the purpose of screwing it into the crankcase from the bottom in holes tapped therein, to hold it in place. The head of the bolt is, of course, relied on when the cylinder casting is bolted to the case.

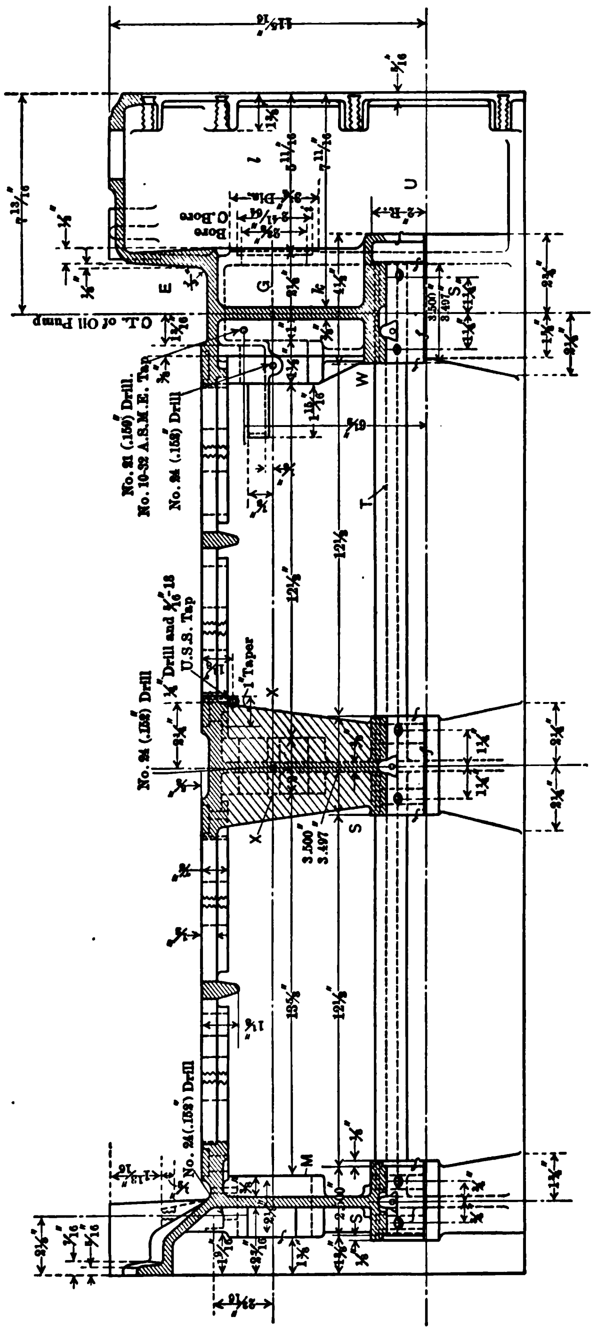


FIG. 98g.—Longitudinal section through crankcase (upper half).

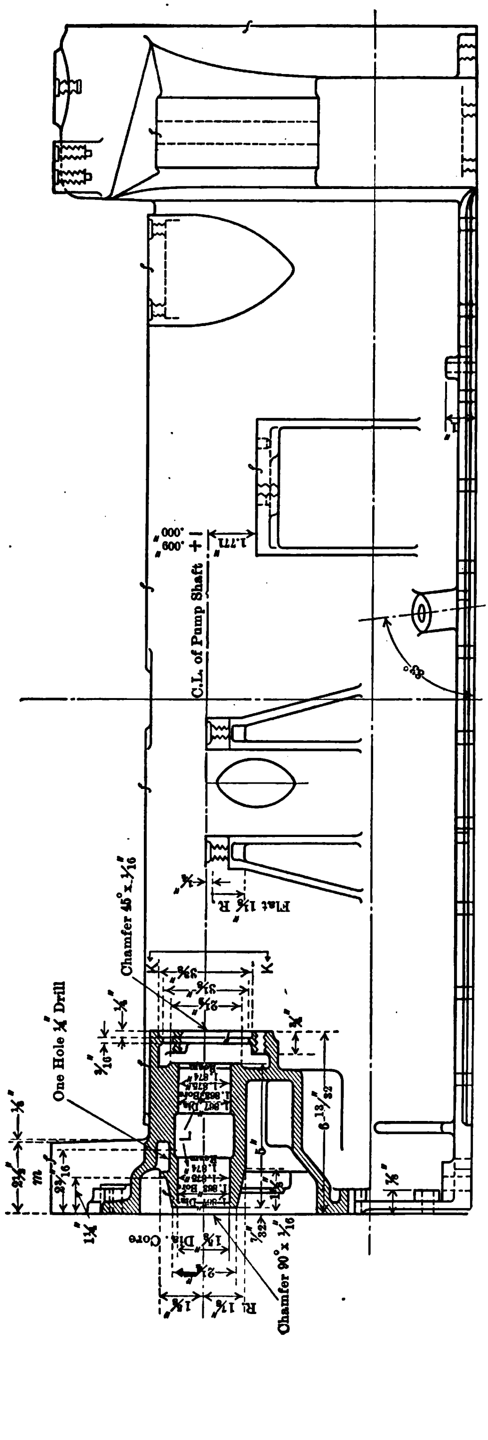


FIG. 98h.—Side elevation (upper half).

The lugs of the aluminum casting which receive the bolts should have a thickness (including thickness of case) of about one and one-half times the diameter of the bolt. The diameter d of the stud or bolt should be as follows: When the cylinders

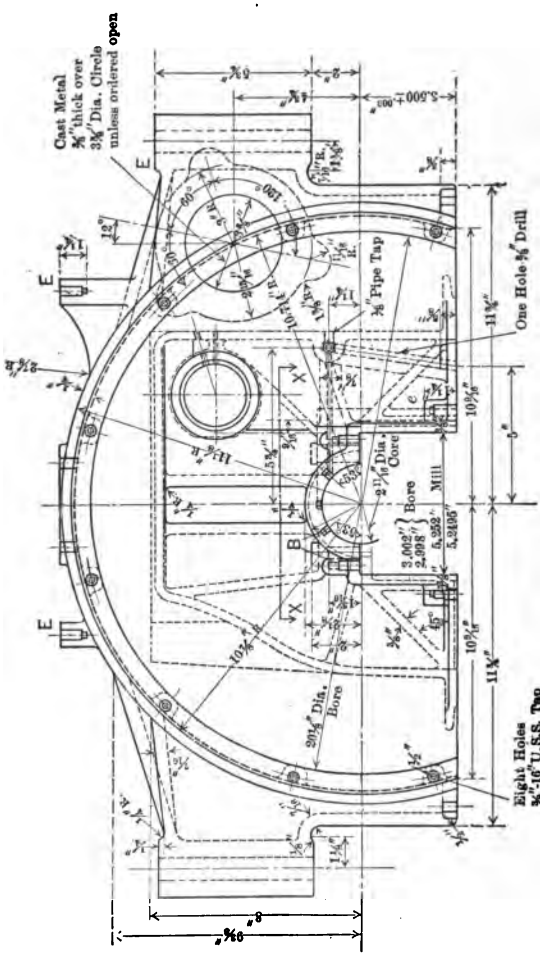


FIG. 98i.—Rear view.

are cast in pairs or en block, and the total number of bolts are from 3 times to $3\frac{1}{2}$ times the number of cylinders, as is usually the case, the diameter d of the studs should be, $d = .125b$ (b being the bore). When the cylinders are cast singly or when 4 studs are used for each cylinder it may be made

$$d = .125b - \frac{1}{16}''$$

There usually are some ribs provided in the crank case to strengthen the lugs, and also partitions in the shape of ribs, between the various cylinders. These partitions extend to the bearings which they support (see Figs. 98*c* and 98*g*). When there is no partition wall, as in the four-throw two-bearing crankshafts stiffening ribs are placed across the underside of the top half of the case.

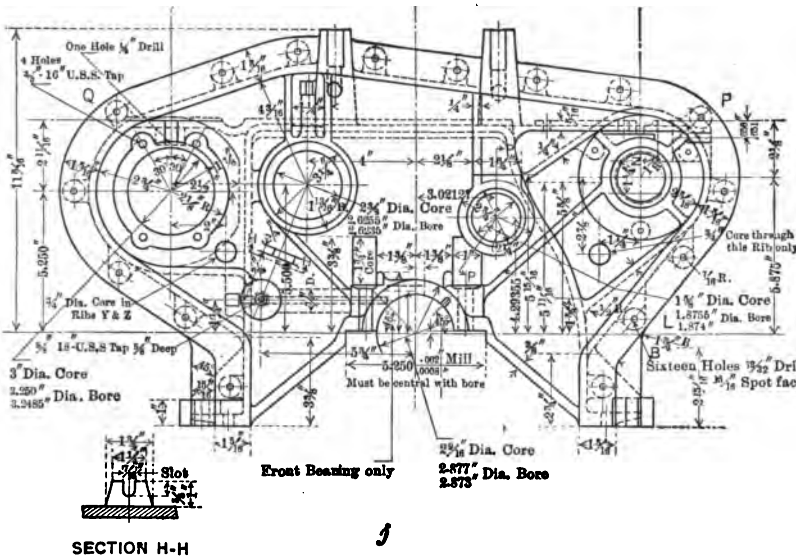


FIG. 98*j*.—Front view.

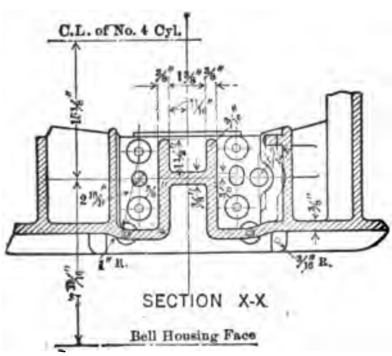
While some portions of the crankcase could be very thin, the foundry experiences difficulties when pouring a case having walls thinner than $5/32$ inch thick; this therefore should be the minimum thickness for even the smallest crankcases. The largest cases are only made about $1/4$ inch thick, and for engines having a bore of from 3 inches to $4\frac{3}{4}$ inches, the average thickness used is $3/16$ inch.

The thickness of the metal upon which the cylinders are resting is made considerably heavier; a simple rule is to make it $\frac{b}{12} + 1/8$, or slightly less if the top is well ribbed between the cylinders, as seen at *r*, Figures 98*c* and 98*e*.

The flanges by which the two halves of the crankcase are bolted together are made from $1\frac{3}{4}$ to 2 times the thickness of

the crankcase metal, the first being used for small engines, and the latter for engines having a bore of $3\frac{1}{2}$ inches or more. The two halves are joined by means of bolts from $\frac{5}{16}$ to $\frac{3}{8}$ inch in thickness, depending on the size of the engine and the spacing of the bolts.

Bolts are spaced from three inches to five inches apart. Usually, when the bolts are close together, no lugs are provided on the flanges for the bolts; if the spacing is more than 4 inches lugs are provided, about $\frac{1}{16}$ to $\frac{1}{8}$ inch thicker than the flanges, and they are spot faced where the bolt heads or nuts are resting on them.



K

Fig 98*k*.—Section X—X.

To make the joints between the flanges oil tight a suitable gasket must be used, and this is made of velumoid, brown paper, or thin cork, and they are preferably shellacked on one side when assembling. Thin felt is not recommended as the oil will ooze through, rendering the exterior sticky and dirty.

Automobile engines are as a rule held in place by three or four supports or supporting arms. When three supports are used (see '*a*' Fig. 62) two are in the rear and one in the front on a cross member or vice versa, two in the front and one in the rear, making a three point suspension. When four arms are provided there are two in the front and two in the rear.

The supporting arms are made of channel sections with the open sides below or above. The former is preferable, for channels open on the top are regular dirt catchers. The supporting arms are resting on the frame and are bolted to it; a piece of wood or a tubular steel spacer being used inside the chassis frame through which the bolts pass, else the frame metal would be bent out of shape when the bolts are tightened.

The metal thickness of the supporting arms varies from $\frac{5}{16}$ inch for light engines, to $\frac{7}{16}$ inch for heavier motors as for instance the class B motor (see Fig. 98*d*), $\frac{3}{8}$ being the average thickness.

Sometimes separate supporting arms of cast steel or drop-

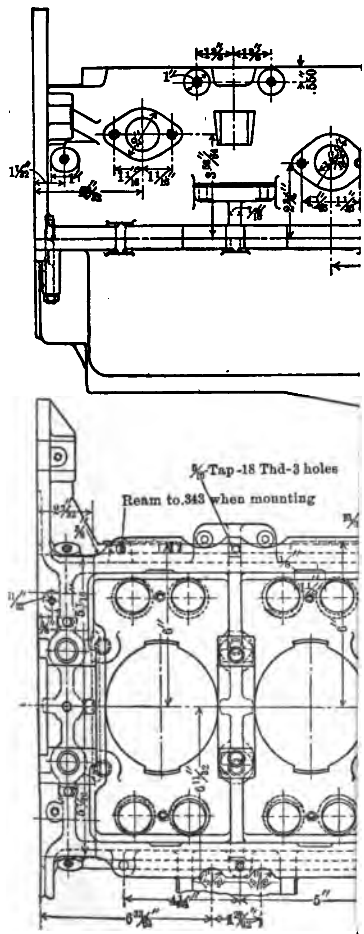


FIG. 99.—Side elevati
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forged steel or of manganese bronze are attached to the crankcase casting; they are fastened to the front and rear end of the crankcase respectively, by means of two large bolts at each end passing through lugs on the end wall of the crankcase, as shown at *D*, Fig. 99*c*.

BREATHERS

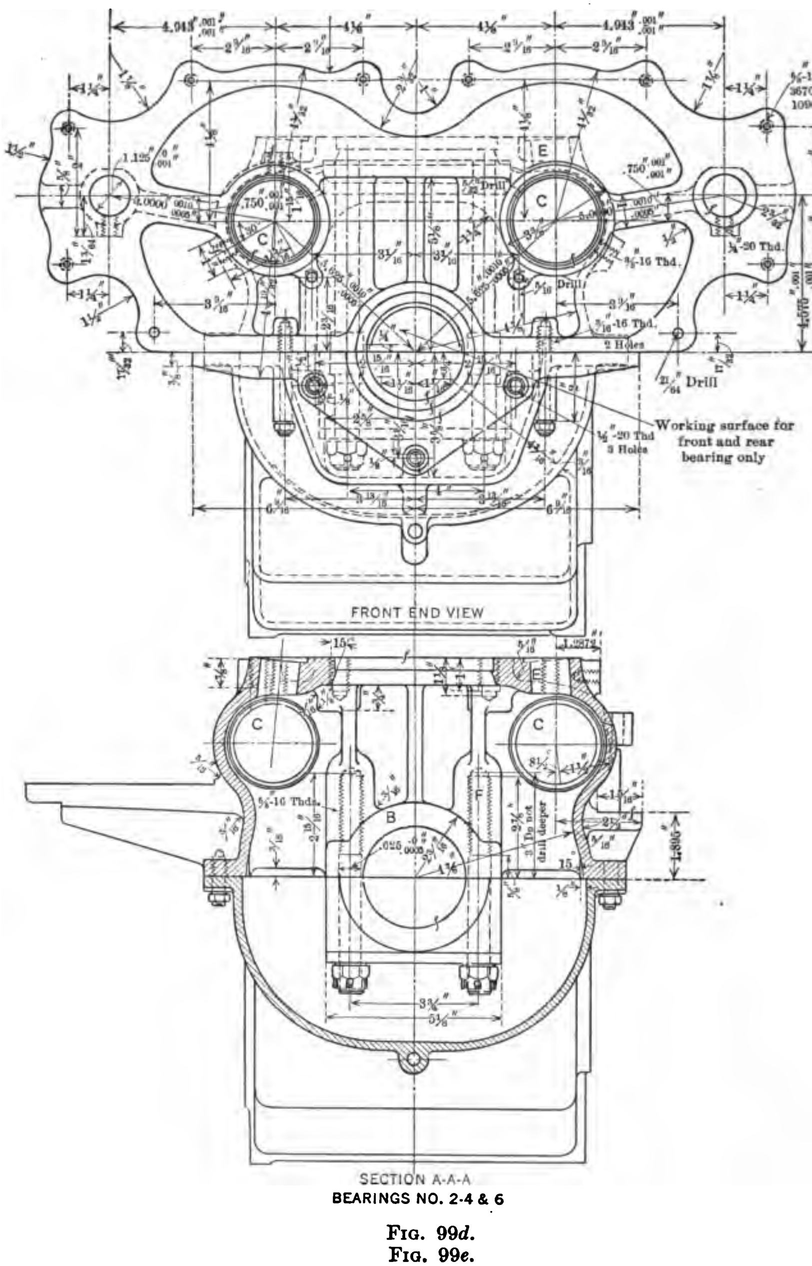
The unequal speed of the pistons during the upper and lower halves of their strokes, and the shanks of the revolving crankshaft set up air currents and an air pressure in the crankcase, which is largely relieved by providing one or two breathers; two are preferable. Pressure in the crankcase will cause oil leaks by forcing the oil through the joints. Even air currents will cause local oil leaks in portions of the crankcase joints. Breathers as a rule have cross partitions, or wire gauze members, sometimes both, to prevent the oil, in the shape of oil globules or in spray, to pass out through the breather, as the oil globules will impinge thereon, condense, and flow back into the case.

BEARINGS

In crankcases the crankshaft bearings are formed in the case, and caps are bolted to the same. The caps, if at the bottom, have to withstand the entire force of the explosion; if on the top, the force of the inertia of the reciprocating parts, and this as we have seen, may be even higher in high speed engines. The caps are held to the bearings by two or four bolts, or studs, preferably four when the bearings are comparatively long. The rear bearing cap which carries the flywheel, should always have four studs or bolts.

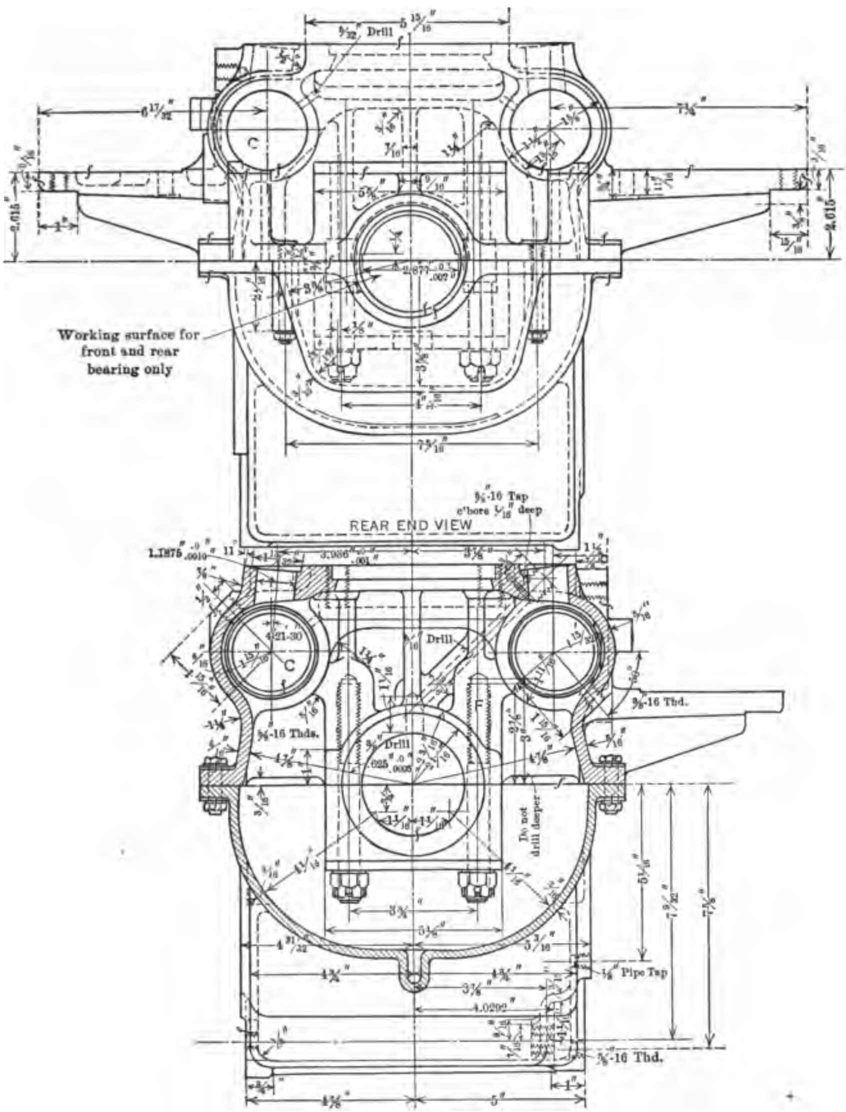
The diameters of the studs for holding down the caps vary from $3/8$ for small engines, to $5/8$ inch for large engines. These studs should be made of nickel steel, on account of the great additional strength for a comparatively small additional cost.

The bearing metal most frequently used for crankshafts is babbitt lined bronze. In former years, sometimes the entire bearing was made of babbitt or anti-friction metals, but on account of their soft nature, the shaft pounded the bearings out of shape, causing play. To-day, the thickness of the babbitt is seldom more than $1/16$ inch thick, the rest being a hard bronze. The bushings are provided with oil holes and oil grooves as done in practice with other bearings, and as will be described in the next chapter.



SECTION A-A-A
BEARINGS NO. 2-4 & 6

FIG. 99*d*.
FIG. 99*e*.



SECTION B-B-B
BEARINGS NO. 3 & 5

FIG. 99*f*.
FIG. 99*g*.

Figure 36*a* shows how the bearing bushings are held in place by screw *x*. Sometimes only studs or other means are provided for preventing the bushings from turning. Figure 98*i* shows the rear end of the case and Fig. 98*j* the front end of the upper crankcase, while 98*k* shows the section *X-X* on Fig. 98*i*. This view discloses the space for the bolts holding the bearing cap and the ribs going to the bearing.

Figures 99 and 99*a* show the aluminum crankcase (side view and bottom view of upper half) of the twenty-four valve, six-cylinder, Pierce Arrow motor previously referred to. As this motor represents the latest product of this well-known concern, it is thought advisable to give complete drawings of the crankcase.

Figures 99*b* and *c* show the top view and the longitudinal section and Figs. 99*d* to *e* the front and rear end views and cross sections.

In these drawings the tunnels provided for the two camshafts are marked *C*.

This crankcase is made of aluminum alloy and attention is drawn to the rigidity of the upper half. The cylinders in this engine are cast in pairs each pair bolted to the case by six $\frac{9}{16}$-inch bolts.

E above the tunnel *C* (Fig. 99*d*) shows the space for the push-rod guide. Note the length of bolts *F* (Fig. 99*e* and *f*), threaded into the upper half of the crankcase to support the lower bearings of the crankshaft. Where sufficient length of thread is provided in the aluminum case the studs may be fastened permanently into the upper half. The castelated nuts *G* (Fig. 99*c*) with split pins are made use of to bolt the lower half of the bearings to the upper half.

CHAPTER XV

LUBRICATION OF ENGINES. TESTING OF OILS

All wearing surfaces must be lubricated to reduce the friction and eliminate unnecessary wear. In the gasoline engine the surfaces or bearings which must be lubricated are: the surface of cylinder wall upon which the piston slides to-and-fro; the crankshaft, camshaft, and connecting rod bearings; the timing gears and the valve operating mechanism, and others.

Various systems have been devised for carrying the oil, or for forcing it, to these wearing surfaces. The oiling system used at

present by most manufacturers is the combination splash and pressure type.

In this system the oil is forced to the main bearings and the crankpin bearings under pressure, at times as high as 40 lb. per square inch, or even higher. The cil works through the ends of the bearings and flows back to the oil sump or oil well at the bottom of the crankcase, which must hold a comparatively large amount of oil.

In this system, the oil is frequently carried to the timing gears at the front end of the case, as a rule the overflow is directed into this compartment. The bottom of the crankcase is provided with receptacles or troughs, into which the dippers or splashers of the connecting rods dip, and splash the oil against the inside of the cylinder wall, the cams, the camshaft bearings (which are frequently provided on the top with oil pockets, to collect oil and lead it to the bearing surfaces) and against the inside of the piston whence the oil finds its way to the piston pin bearings.

The oil level in the troughs is kept constant, either by the height of the troughs, by overflow stand-pipes, or by holes in the sides of the troughs, through which the surplus oil, which constantly accumulates therein, flows to the sump below. Sometimes no splashers are used, the oil spray in the crankcase being the lubricating medium, as will be mentioned later.

In the purely "Splash Lubrication" system, the main bearings as well as the cylinders and other parts of the engine are lubricated by the splash alone, while in the full pressure-feed system all the lubricating surfaces, piston-pins, cam-gearing, and the main bearings are lubricated by oil under pressure. In the combination splash-pressure system the cylinders and the piston-pins are lubricated by the splash. One of the disadvantages of the splash system has been an excessive lubrication of the cylinder walls, which meant that some of the oil was carried into the combustion chamber, causing a smoky exhaust and a more rapid formation of carbon in the cylinders. This difficulty has been largely overcome by means of scraper rings at the bottom end of the piston as mentioned in the chapter on "Pistons." These scraper rings scrape off a portion of the oil splashed onto the cylinder walls. Furthermore, by improving the fit and reducing the clearance between cylinder and piston, less oil can get into the cylinder. With high-speed engines where the oil is fed to the bearings under pressure, the oil also has a

cooling effect on the bearings thus keeping down their temperature. There seems to be a tendency now to make provisions for carrying the heat away from the oil as it is found that excessive crankcase temperatures are injurious, since high temperatures impair the lubricating qualities of the oil.

Some manufacturers provide ribs or radiating fins at the bottom of the crankcase, containing the oil trough, others use thin sheet stamping for the troughs with the same end in view.

It is important that there should be provided neither too much nor too little oil for the cylinders, for as stated before, when the cylinders are over-oiled there will be smoke; when they are under-oiled, especially with the splash system, there may not be sufficient lubrication for the piston-pin bearings, the crank-pin, and the crankshaft bearings.

A pocket or cavity should be provided at the lowest part of the crankcase sump where all sediment and all foreign matter will settle and not reenter the circulating pump (see *M*, Fig. 36). The said cavity should be lower than the level at which the oil is carried off by the pump. Sometimes small partitions are placed between the pump and the rest of the oil reservoir. Fine mesh metallic screens are usually fitted between the main crankcase and oil sump below, or over the inlet side of the circulating pump, so as to positively prevent foreign matter like metallic sediment, cotter pins, etc., from entering the oil circulation. Figure 34 shows the screen used for filtering the oil in the Class B Military Truck Engine, while *S*, Figs. 35 and 36, show the screens of the Haynes and Waukesha motors previously referred to.

To obtain a maximum efficiency it is necessary that the engine be not over-oiled, when run at moderate speeds with light loads, and that it obtain sufficient oil when run at relatively high speeds and developing large power, and that the temperature of the crankshaft and the crank-pin rise not beyond the limit at which the oil retains its lubricating qualities.

As a rule engine-oiling systems are arranged so that an increase in engine speed will automatically increase the amount of lubrication. This, however, is not quite satisfactory, for the reason that a car driven on a level smooth road requires very much less power than when driven over bad roads or up an incline. From this it is apparent that proportioning the lubrication to the engine speed may cause under-oiling under some conditions or over-oiling under others. There seems to be a tendency now to con-

nect the oil control with the engine throttle, so that when the throttle is opened to increase the torque output of the engine the amount of oil fed to the bearings is increased. Some makers using the splash, while others, the force-feed system, have successfully made such connection between lubricating system and throttle. If used with the force-feed system, it will permit higher pressure under which the oil is forced to the bearing surfaces; in the splash system, it may allow a higher level of the oil at the bottom of the crankcase.

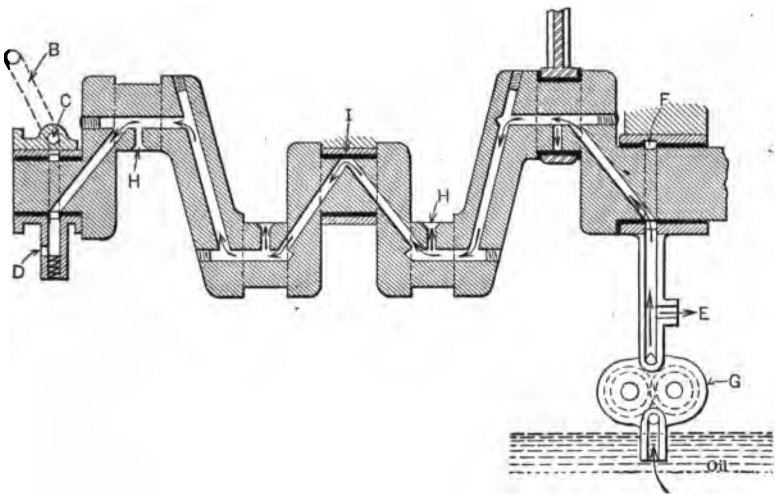


FIG. 100.—Force-feed lubrication to force oil to main bearings and crank-pin bearings.

Figure 100[1] shows a method for forcing the oil to the crankshaft and crank-pin bearings. It can easily be seen that by interconnecting in some manner the arm *B* with the outlet valve *C* at the end of the oiling system, the pressure in the oil flow can be increased or diminished. In other words when the throttle is wide open the arm *B* will entirely close the outlet valve *C* which would cause the pressure to rise to a predetermined point, as regulated by a spring control of the pressure outlet *D*. At *E* the pipe will flow to the pressure gauge on the dash, and in this system at light loads the pressure will be very low, increasing as the throttle is opened. As seen from the drawing, the oil reaches

[1] See "Problems in High-speed Engine Design," by A. P. Brush, S. A. E. Transaction, Vol. II, 1916.

every bearing surface on the crankshaft. At *F* there is a groove all around the bearing or only for some distance around the bearing) which is usually provided in some form in pressure feed systems) which is kept continually filled by the pump *G*, and from this groove the oil flows through the crankshaft in the direction of the arrows. *H* indicates holes for the flow of oil to the crank-pin bearings and to the center crankshaft bearing at *I*. In the Hudson high-speed motor the amount of oil flowing through the pump is increased or decreased by an eccentric connected with the carburetor throttle. As this eccentric is operated it holds the plunger of the oil pump away from the cam which actuates it at low speeds, but increases the plunger motion when the motor speed is increased. In other words, the oil pump has a short stroke when the motor is idling or running at light loads, but as the throttle is opened the pump plunger stroke is increased in proportion.

In most of the pressure lubricating systems very little pressure is required for proper lubrication of the engine when it is running under light loads, but when more power is required more oil must be forced to the bea rings; this of course requires a higher oil pressure. When the force-feed system is used throughout, the oil from the crank-pin bearing travels up the connecting-rod in a small pipe (or in a hole drilled in the connecting-rod), to the piston-pin bearing, but with the splash-pressure system the cylinders and the piston-pins are lubricated from the splash and from the oil flowing out of the crank-pin bearings and thrown against the cylinder walls and to the inside of the piston by centrifugal force. In some designs no splasher is used at the bottom of the connecting-rod, the cylinder wall being lubricated by the oil-spray or mist arising from the oil forced through the crank-pin bearings. In high-speed engines the maximum crank-pin bearing pressure occurs on the inner side of the crank-pin on account of the inertia and centrifugal forces set up by the piston and part of the connecting-rod as was mentioned in preceding chapters. It is always advantageous that the oil should be fed to a bearing at some distance from the point of maximum pressure as this will leave more room for the oil to enter the bearing, thus improving the lubrication.

In pressure-feed systems as a rule gear oil-pumps or plunger pumps are used. The gear oil-pump consists of a casing in which two spur gears are snugly fitted, one is driven by means of its

shaft and the other by being in mesh with the former. The oil enters the housing on the side on which the meshing teeth separate and fills the spaces between adjacent teeth and the wall of the housing. The oil is carried around with the teeth to the opposite side of the housing where it is forced through the outlet. Figure 101 shows the gear pump arranged in the Hercules Engine. *N*, Fig. 36*a* shows the gear pump in the Waukesha Motor and how same is driven through a gear *O* attached to the camshaft. In the plunger type of pump, there is simply a pipe or cylinder with a plunger on the inside; the plunger as a rule being forced out

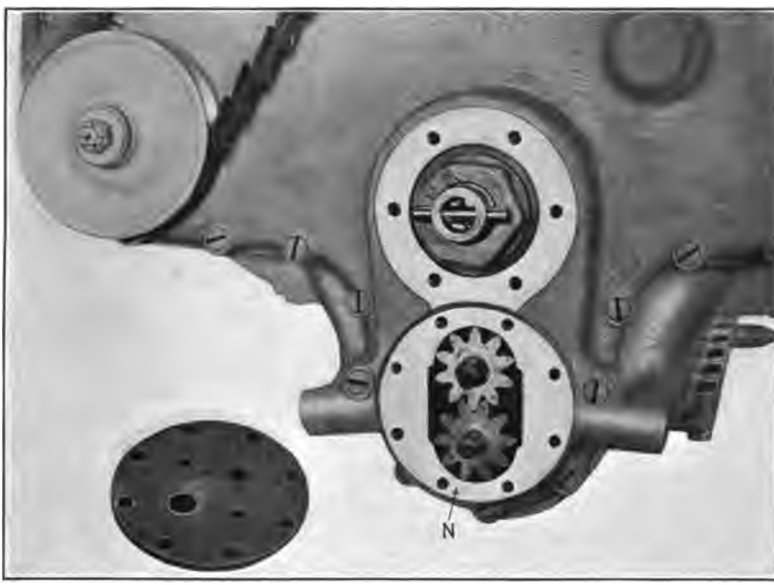


Fig. 101.—Gear oil-pump of the Hercules engine.

again by a spring. The plungers frequently are operated by an eccentric provided for this purpose on the cam shaft.

The trucks in war service have demonstrated that for hard duty, pressure lubrication is necessary to all the bearings of the crankshaft, crank-pin and camshaft bearings. With the floating bushing in the upper end of the connecting rod and a floating pin, pressure lubrication here has not been found necessary.

Of late years the crankshaft has been continually strengthened to obtain rigidity, which, in turn, made necessary the increase of the crankshaft diameter. This results in a higher linear speed of

the bearing surface, which in turn, necessitates better lubrication. For this reason the number of models using the pressure feed systems have continually increased. When the linear speed of the bearings exceeds 1000 feet per minute, it is most important to maintain a film of oil between the bearing surfaces, hence pressure feed for such bearings in high-speed engines is essential. Working pressure of the oil in high-speed engines is about 35 lb.; for ordinary touring-car engines about 20 lb. seems to be the average. In all pressure feed systems it is important that the strainers be of ample size so that when portions of them become clogged up, they do not prevent the flow of the amount of oil required.

LUBRICATION OF CLASS B MILITARY TRUCK ENGINE

In the Class B engine, the gear pump *N* is located at the end of the crankcase as seen in Fig. 34. The oil is forced through a passage drilled in the body of the pump to a header, extending the full length of the crankcase, as seen in Fig. 98*b*. From here the oil is forced through drilled passages to the main bearings. Grooves are provided in the main bearings which are in constant communication with the oil holes drilled through the crankshaft to the connecting-rod bearings (see the crankshaft, Fig. 84). Oil tubes on the connecting-rods lead the oil to the piston-pins. The relieve valve Fig. 34*a*, section AA (which in this case is set for a pressure of 10 lb.), is fitted to the front end of the oil header, and the over-flow from this valve lubricates the timing gears, and thereby the other gears in the front of the motor. The oil pump is supported from the upper crankcase, and extends down into the oil sump. No parts whatever outside of the settling chamber and the larger strainer are attached to the lower half of the crankcase, in consequence this can be removed without interfering with anything else.

Figure 102 shows a cross section of the Buda HU Motor, which was adapted for the "Model A" United States Government Truck (1½ ton). In this figure the strainer *S* and oil pump *N* are seen at the bottom of the crankcase. The arrows indicate the direction of the oil and show how it is forced to the main pipe which extends through the whole length of the crankcase. The main pipe lubricates the three main crankshaft bearings and the camshaft bearings. From the crank-pins pipes lead the oil to

the upper connecting-rod bearing. The over-flow and oil thrown off by centrifugal force from the crank-pins lubricates the cams and the cylinder walls.

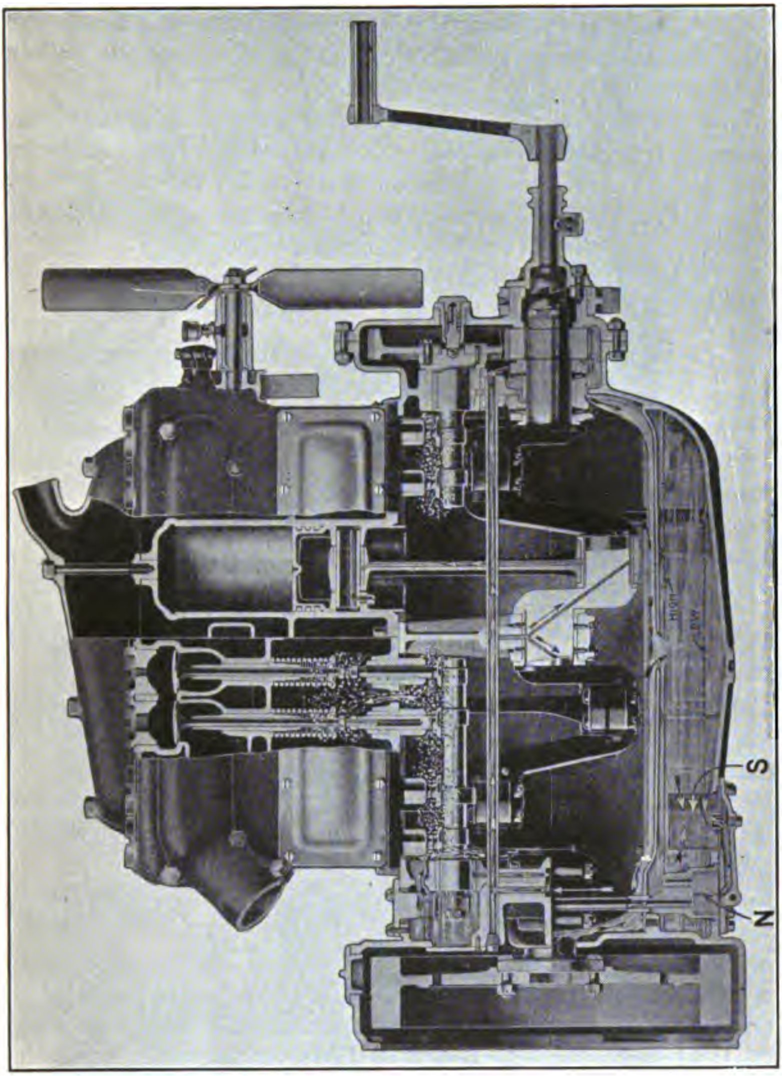


Fig. 102.—Sectional view of Buda engine.

Figure 103 shows the pressure regulator of this engine. The pressure can be altered by adding or removing washers underneath the screw *s*, which bears on a spiral spring as seen.

In some lubricating systems, as in the Packard Twin-six, a hollow camshaft is used for the lubrication of the camshaft bearings, as seen in Fig. 104. Here *H* is the oil header extending through the entire length of the case. From *H* the oil flows through leads *L* to the main crankshaft bearings. From the front crankshaft bearing lead *m* conducts the oil to the hollow camshaft *c*.

Note the ribs *R* at the front and rear ends of the crankshaft before it leaves the crankcase. They return to the crankcase the surplus oil, by passage *H* (Fig. 99*c*) which flows through the ends of the crankcase bearings.

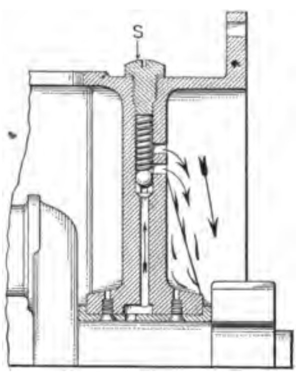


Fig. 103.—Pressure regulator of HU Buda engine.

TESTING OIL

Oils used in automobile engines are as a rule subjected to the following tests:

1. Heat test or quality test.
2. Flash test.
3. Viscosity test.
4. Cold test.
5. Acidity test.

1. *Heat Test or Quality Test.*—In this test the sample of oil is heated slowly in a flask until yellow vapors appear. Then the temperature is held constant for fifteen (15) minutes, after which the flask is set aside for twenty-four (24) hours. Good oil will darken in color but remain clear and no sediment will settle. Bad oil will turn jet black and leave a black sediment at the bottom which proves the presence of sulphuric acid or sulpho compounds in the oil. Such oil should be rejected.

2. *Flash and Fire Test.*—Flash and fire tests may be made with the Cleveland open cup flash and fire tester. Heat should be applied so that the temperature of the oil increase at the rate of about 10°F. per minute. After the temperature has risen to about 300°, a small flame, either a small wax candle or a gas flame, should be quickly moved to the top of the cup in which the oil is heated, and again quickly withdrawn, at every 5° rise in temperature, until the vapor arising from the oil will ignite and go out again. The temperature at which the vapors ignite is called the *flash point* of the oil. If the oil is heated still more

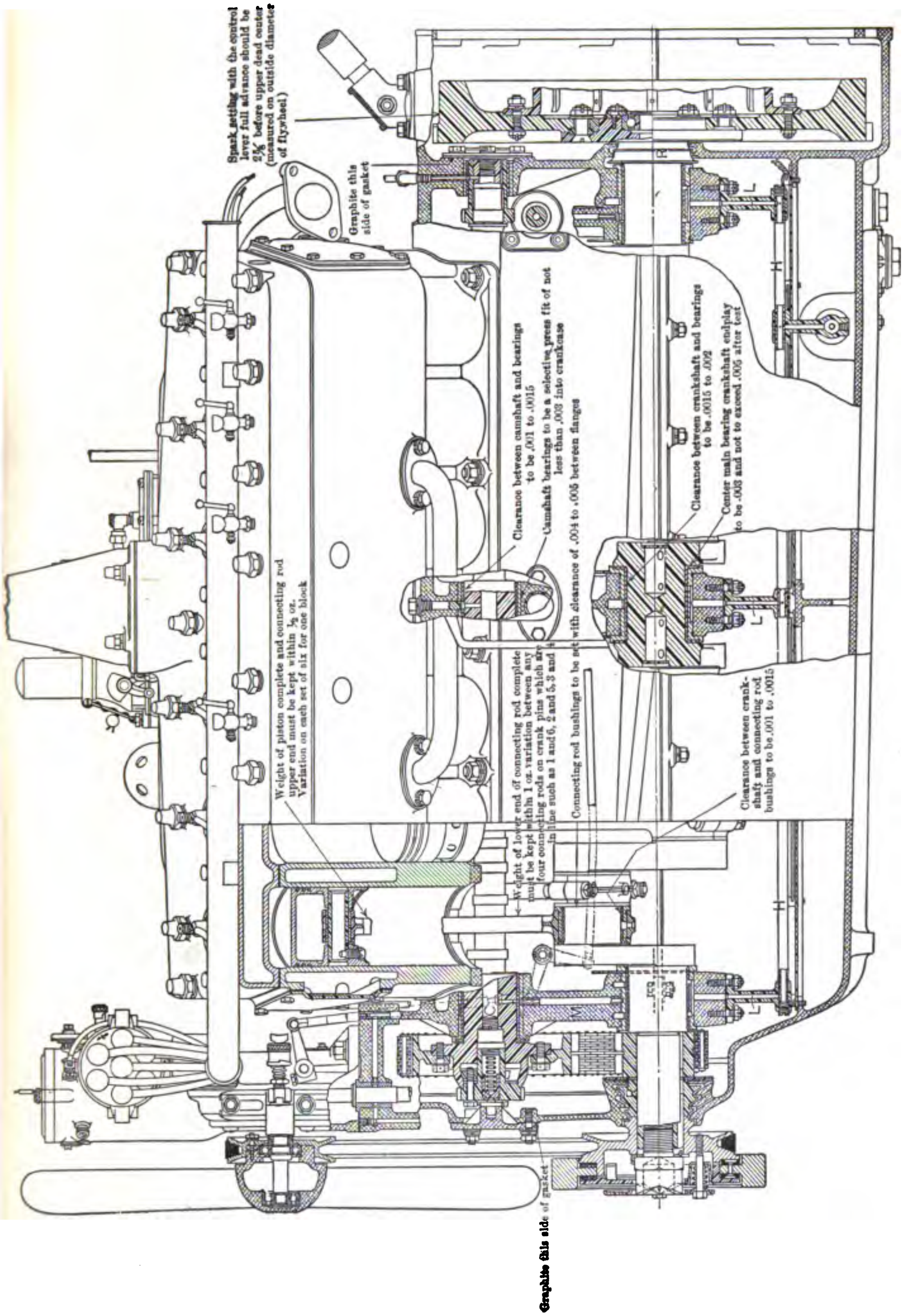


FIG. 104.—Packard twin-six engine.

until the oil itself will take fire when a light is brought to it, the temperature at which the oil will burn is called the *fire test* of the oil.

Another instrument for testing the flash point of an oil is the Pensky-Martin oil tester which is considerable more expensive than the Cleveland tester, but tests can be made more easily and accurately.

3. *Viscosity Test.*—The viscosity (or cohesion) of an oil is its thickness, so to say, and is usually tested by Saybolt's viscosimeter. In the viscosity test, the oil, after being heated to a certain temperature, is permitted to run through a small hole or orifice, and the *Number* of *Seconds* required to fill a flask having a volume of sixty (60) cubic centimeters is called the viscosity of the oil. The temperature at which the cylinder and engine oils are usually tested is 210°F. of the oil to be tested, while the temperature of the bath in which the oil to be tested is heated is 212°. Frequently the viscosity of oil is tested at 100°F. When heated, all oils become thinner, that is to say, they lose in viscosity.

4. *Cold Test.*—In the cold test a straight round bottle about 1½ inches in diameter is half filled with oil and a thermometer inserted into the oil through the cork of the bottle. The bottle is wrapped in a blotting paper and then placed in a mixture of chunks of ice and salt. At every few degrees Fahrenheit drop in temperature, the bottle should be tilted to see if the oil will pour. The temperature at which the oil will not move at all when the bottle is kept tilted for ten (10) seconds, is called the *cold pour test* of the oil. Sometimes 5° is added to the temperature when noting its pour test. For instance: Suppose an oil will not pour at all at 10°F. Then adding 5° to it, the *pour test* of the oil will be denoted as being 15°F. In addition to the pour test the *cloud test* is often made use of which shows the temperature at which the oil first begins to get cloudy when subjected to the cold test.

5. *Acidity Test.*—In addition to the heat or quality test described above, another simple test for determining the presence of acid can be made by washing a small sample of the oil to be tested in warm water and then pouring the water off and testing it with a small piece of neutral litmus paper. If there is acid in the water, the litmus paper will turn pink. When such is the case, the oil is liable to cause corrosion of the shafts or bearings to be lubricated.

CHAPTER XVI

OFFSET CYLINDERS

In Chapter IX it was shown that there is a certain side thrust against the cylinder wall which causes friction between the piston and the cylinder. It was shown that the pressure with which the piston bears against the sides of the cylinder wall at any particular instant = $P \times$ tangent A, where A is the angel formed by the connecting-rod with respect to the axis of the cylinder (see Fig. 71) and P is the total pressure in pounds on the piston head. Evidently the friction between cylinder wall and piston will be greatest on the power stroke when the pressure on the piston head is greatest.

In order to reduce the side thrust, the angle A, during the power stroke would have to be reduced, and this is done by offsetting the crankshaft center line from the cylinder axis toward the side (opposite to that on which the crankpin travels on the power stroke) so that the connecting rod will move down during the power stroke in a more vertical line. The result will be a smaller angle during the power stroke but a greater angle on the return stroke of the piston, and since the pressure on the piston head is much greater during the power stroke than during the compression stroke, by somewhat offsetting the center of the crankshaft with respect to the center of the cylinder, a point can be reached when the total friction or side thrust during the power stroke equals the side thrust during the compression stroke.

With an offset crankshaft the piston is not entirely at the ends of its stroke when the crankarm is in its vertical position, but a little later, when the connecting rod and the crankarm form a straight line; neither are the dead centers of the piston exactly 180 degrees apart, but the angle is slightly less on one side, and slightly greater on the other. It is recommended to students interested in this problem, to draw diagrams with various offsets and various lengths of connecting rods and study the results.

One point which must be considered when designing an engine with an offset crankshaft and a given piston stroke, is that the

crank of the crankshaft must be slightly shorter than one-half the piston stroke. The actual difference can easily be determined by laying it out graphically on the drafting board.

The distance of the amount of the offset (distance between crankshaft and cylinder center lines) varies from about one-fourth to one-sixth the bore. As mentioned before, when the cylinders are offset, the angle *A* is smaller throughout the power stroke and larger during the compression stroke.

Offset cylinders are especially useful with motors having short connecting-rods, for the longer the connecting-rod the less the side thrust.

CHAPTER XVII

MANIFOLDS

INTAKE MANIFOLDS

The intake ports of an engine are connected with the carburetor by a lead or pipe, having as a rule branches for each pair or for each group of cylinders, and this pipe is called the intake manifold, or the inlet manifold.

In the design of the intake manifold there are two chief considerations: first, each cylinder should receive the same quantity of charge; and secondly, the mixture should be equally well vaporized when entering each cylinder. With the fuels used to-day, a great portion of it passes through the manifold as a mist suspended in the air current. A certain velocity is required to maintain this suspension of fuel particles, depending upon the size of these liquid particles. When the speed of the air current is decreased below a certain velocity the larger particles of the fuel condense. The gas velocity in the manifold depends upon the engine speed, and upon the area of the intake manifold. It is the variable engine speed which makes the design of a correct manifold more difficult, for, if the manifold is too small there will be a loss in volumetric efficiency at high speeds; on the other hand, the larger the area of the manifold, the less the velocity of the gas and therefore if the engine moves very slowly as in starting, the fuel mist cannot be carried to the cylinder as a mist, and this renders the starting of the engine difficult.

The speed in the manifold depends also on the friction between the gas and the interior surface of the manifold, called the "skin-friction;" therefore, the intake manifold should be smooth as the skin friction reduces the speed of the portion of the charge which touches the walls. This skin friction will wet all the manifold wall surfaces, and form a liquid film, but the latter is subjected to the moving air column which tends to evaporate the liquid; this takes place slowly, however, since evaporization can only take place from the surface. It shows the importance of eliminating pockets where any depth of liquid can accumulate; instead it should be endeavored to increase the surface of the liquid and for this reason it is often suggested that the interior of the walls be roughened.

Another difficulty to proper distribution of the mixture is the tendency of any liquid or gas in motion to seek the periphery of all curves. As the particles of liquid are greater than air they are thrown against the outside of the curve by centrifugal force, thereby lessening its velocity by the impact, and depositing particles of liquid. Naturally, the smaller the radius of the curve, the greater the tendency to cause deposition of these particles.

When an engine is hot, the heat will assist in the evaporation of the liquid, but it is hard to start when cold. If the bends in the manifold are properly designed they may even assist in the more thorough mixing of the gases especially if heat is applied to the manifold. (Some manifolds with bends have been enormously improved when properly water jacketed and they may give even better results than others without bends.) The efficiency of the manifold will also depend upon the carburetor used, for a highly atomizing carburetor will act differently from an ordinary carburetor. Since bends cause a deposition of particles of fuel, the drainage of these bends should be directed toward the heated surfaces and should be drained to slope away from the cylinders.

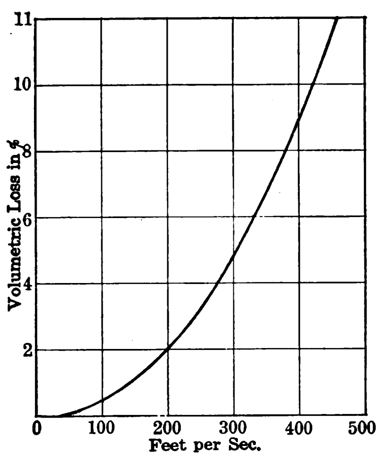


FIG. 105.

If the manifold walls are kept at a temperature equal to or above the average boiling point of the fuel (about 180°F. with present-day gasoline) the walls will not become wet (for this reason many manufacturers water jacket the intake manifold) but when the temperature is lower, the particles of fuel, when deposited, are carried away by the air stream and may enter the valve chambers in liquid form. This will cause difficulties in uniform distribution. Superior results are often obtained with jacketed manifold when the air supply is heated, as thereby heat is furnished to the atomized charge while the fuel particles are still in suspension. The heated air may cause a loss in volumetric efficiency but this is outweighed to a great extent

by the increased efficiency arising from a more thorough evaporization, and therefore, instead of a loss in power, frequently a gain is realized. All this, however, largely depends on the general design of the engine, for what might give excellent results in one design has often been found wanting in another.

Figure 105[1] is a table giving the volumetric loss of motors at different velocities of the gas, in feet per second. As stated by Mr. Brown in the paper referred to:

"A liquid fuel engine of average flexibility requires at least from 12 to 15 times the minimum amount of air at maximum speed. If a velocity of, say, 30 feet per second is necessary to maintain suspension of fuel atomized to a given fineness, and if the area of the manifold is such that this velocity is to be maintained at the lowest speed, then at maximum speed the velocity would approach 450 feet per second. This would entail a loss of volumetric efficiency, from velocity head alone, of over 11 per cent., without considering friction which would increase this loss."

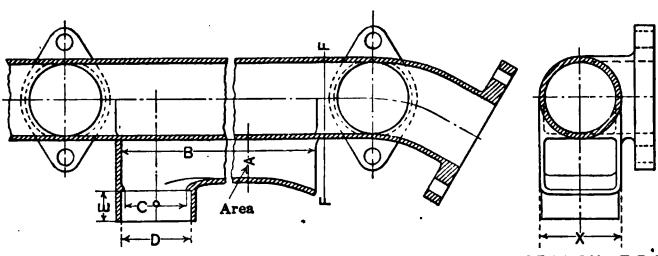


Fig. 106.—S. A. E. standard exhaust manifold air heater.

Of course, in practice at the lowest speed the velocity in manifolds need not be 30 feet per second, or 1800 feet per minute, therefore the loss will not be as great at high speeds. Ordinarily, the area of the manifold is such that at the maximum horsepower, the gas speed through the manifold is from 9000 to 12,000 feet per minute. This, of course, will vary in different engines, depending upon the general design.

To obtain a more thorough vaporization of the fuel, for years, many manufacturers used a heated air intake, by attaching a pipe to the carburetor and the outside of the exhaust pipe or manifold (see Fig. 106).

[1] See S. A. E. Transactions, Vol. 9, Part I, by A. B. Brown.

In some designs, as in the Hercules Engine (see Fig. 32), the intake passage is cast integral with the cylinder block and is submerged in the water jacket. The section of the intake passage is well rounded at every corner so as to retard the gas velocity as little as possible. In this manner only a short external intake manifold is required to the carburetor, and the gas, passing from one side of the cylinder to the other (thus in a well-heated passage), assists the carburation of the fuel. By having the section of this intake passage of the right area a high gas velocity can be obtained, insuring an efficient suspension of fuel particles in the mixture as it rushes into the cylinders.

Figure 106 shows the design of air heaters recommended by the Society of Automotive Engineers with corresponding dimensions for given size carburetors.

TABLE X

Nominal carburetor size	Diameter carburetor outlet	A sq. in.	B	C	D	E
1/2	11/16	0.56	3 1/2	13/16	1	5/8
5/8	13/16	0.79	4 1/8	15/16	1 1/8	5/8
3/4	15/16	1.04	4 3/4	1 1/16	1 1/4	5/8
7/8	1 1/16	1.33	5 3/8	1 3/16	1 3/8	5/8
1	1 3/16	1.66	6	1 5/16	1 1/2	3/4
1 1/4	1 7/16	2.43	7 1/4	1 5/8	1 11/16	3/4
1 1/2	1 11/16	3.35	8 1/2	1 7/8	2 1/16	7/8
1 3/4	1 15/16	4.42	9 3/4	2 3/16	2 3/8	7/8
2	2 3/16	5.63	11	2 7/16	2 5/8	7/8
2 1/2	2 11/16	8.5	13 1/2	3	3 3/16	1
3	3 3/16	12.0	16	3 9/16	3 3/4	1
3 1/2	3 11/16	16.0	18 1/2	4 1/8	4 5/16	1

"The mouth or inlet should be flared to give a stream-line effect and when possible it should point rearward from the radiator. The corners at the outlet end of the passage should also be curved as shown, to minimize resistance to the air current. The dimension X should be not less than the outside diameter of the exhaust pipe. The outer edge of bore D should be slightly rounded. When only a portion of the air supply is to be preheated the diameter D should be bushed to the desired size.

"Carburetor air heaters differing from the type shown, whether integral with the exhaust pipe or separate attachments, should have heating surfaces and areas at least equal to those of the type

shown, and the heater outlet dimensions *C*, *D*, and *E* should agree with the figures given in the accompanying table."

Figure 107 shows the dimensions of flanges recommended by the Society of Automotive Engineers, between the carburetor and the intake manifold.

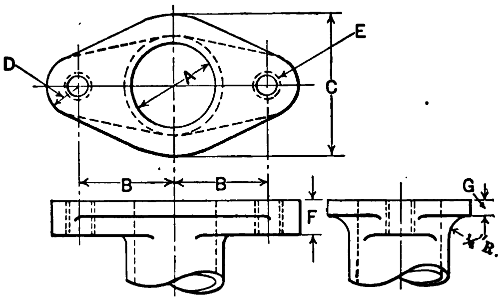


Fig. 107.—Manifold carburetor flange.

TABLE XI

Nominal carburetor size	A	B	C	D	E[1]	F	G
1/2	11/16	27/32	1 1/4	9/32	1/4×20	5/16	3/32
5/8	13/16	29/32	1 7/16	9/32	1/4×20	5/16	3/32
3/4	15/16	1 1/16	1 5/8	11/32	5/16×18	3/8	1/8
7/8	1 1/16	1 1/8	1 3/4	11/32	5/16×18	3/8	1/8
1	1 3/16	1 3/16	1 7/8	11/32	5/16×18	13/32	5/32
1 1/4	1 7/16	1 11/32	2 3/16	13/32	3/8×16	15/32	3/16
1 1/2	1 11/16	1 15/32	2 1/2	13/32	3/8×16	15/32	3/16
1 3/4	1 11/16	1 21/32	2 13/16	15/32	7/16×14	9/16	7/32
2	2 3/16	1 25/32	3 1/8	15/32	7/16×14	9/16	7/32

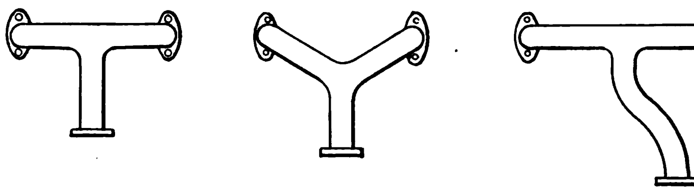
[1] U. S. Standard thread. All dimensions in inches.

SIDE-OUTLET CARBURETOR FLANGES

The same standard flange dimensions should be used with the long diameter of the flange in a *vertical* plane for the attachment of carburetors of the side-outlet type.

Figures 108, 108*a*, and 108*b* show the three most common types of intake manifolds used on four-cylinder engines. Note that in these manifolds there is no enlargement of the cross-sectional area (which would tend to retard the gas velocity). There are no pockets for any liquid fuel to be retained anywhere, but the

slopes are all toward the central member. The length between any cylinder flange and the carburetor is the same, thereby insuring an equal quantity of charge in all the cylinders. Note the large radius curves in the central member of Fig. 108*b*, which is the type of manifold used on some very successful cars. Figure 109 shows that of a six-cylinder engine, and discloses how the passage between carburetor flange and each cylinder is made of equal length.



FIGS. 108, 108*a* AND 108*b*.—Intake manifolds.

Each intake cylinder flange supplies two inlet ports. This is shown on Fig. 110, which is a cross section of a pair of cylinders of the Mack Engine, the section being taken on the line *C-D* of Fig. 38*a* and line *E-F* of Fig. 38. Note that the exhaust ports *E*, after being contracted somewhat when leaving the valve (see Fig. 38*a*) enlarge toward the outside of the cylinder casting, and each exhaust port *E* (see Fig. 38*a* and Fig. 110) discharges directly to the outside, individually, while the inlet ports *I*

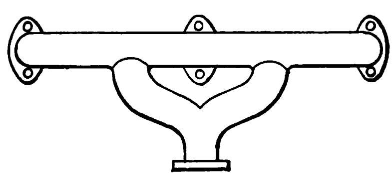


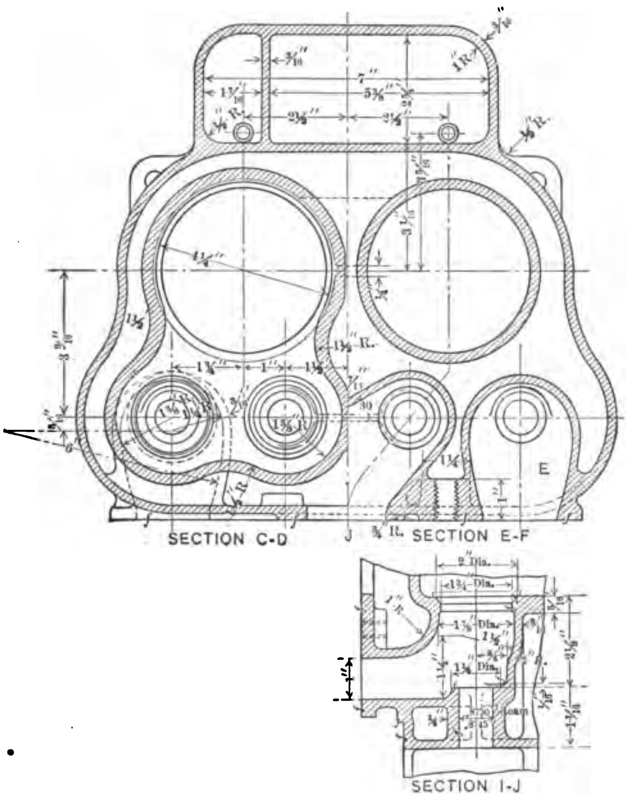
FIG. 109.—Intake manifold for 6-cylinder engine.

of two adjacent cylinders are joined together. (In some designs the center exhaust ports are likewise joined.) Figure 110*a* shows the cross section of the inlet valve port on line *I-J*, Fig. 38 shows the attachment of the manifolds to the cylinder when both the inlet and the exhauxt manifolds are on the same side of the cylinder.

Figure 35 shows the inlet manifold attached to one side of the cylinder and the exhaust to the other side. There is a very short manifold between the carburetor and the cylinder casting,

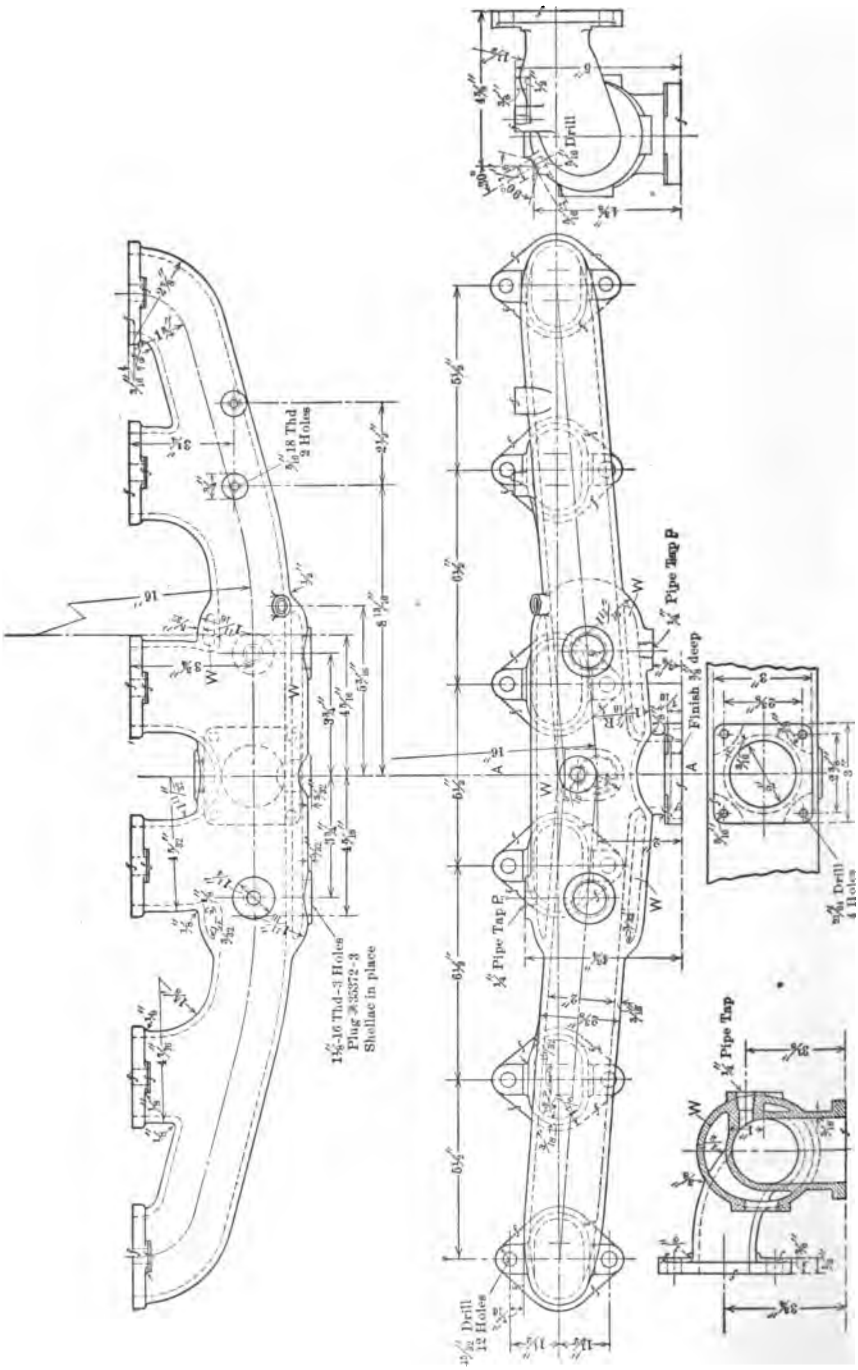
the inlet passage being extended between the cylinders to the opposite side (as shown in dotted lines), where the valves are located. In addition to heating the fresh charge after it leaves the carburetor there is a hot-air intake *h* as shown. With low-grade fuel this would certainly give the highest thermal efficiency.

A very interesting design is that shown in Figs. 111 and 111*a*, which is the water-jacketed intake manifold of the twenty-four



FIGS. 110 AND 110*a*.—Cross section of a pair of cylinders of Mack engine.

valve Pierce Arrow motor. This manifold is made of cast aluminum water jacketed around the central portion as shown at *W*, P_1 and P_2 show the tapped holes for attaching the water inlet and outlet pipes. The water enters at P_1 from the cylinder water jacket and leaves at P_2 which is connected to the water outlet on the top of the cylinders.



FIGS. 111 AND 111a.—Intake manifold of twenty-four valve, six-cylinder Pierce Arrow motor.

EXHAUST MANIFOLDS

The object of the exhaust manifold is to carry the burned gases from the cylinder to the muffler, whence they escape to the atmosphere. The manifold should have sufficient area to cause very little back pressure; in fact, it should be large enough that the gases may expand after they leave the exhaust valve port, thereby lowering its temperature. It is important to avoid over-heating the cylinder casting, and this can be accomplished in two ways—lowering the temperature of the exhaust gas or increasing its speed through the manifold; the quicker the discharge is expelled the less heat can it give up to the cylinder casting. To reduce the temperature the exhaust manifold is frequently ribbed by fins on the outside in order to increase the radiating surface.

Regarding the speed of the exhaust it has been found that when the exhaust port area is contracted, soon after it leaves the exhaust valve, and is then again increased toward the flange,

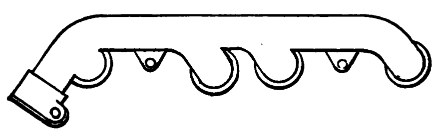


FIG. 112.—Exhaust manifold for 4-cylinder engine.

where the exhaust manifold is attached, more power can be obtained from the engine, as the speed of the exhaust gas is increased; this tends toward more power on account of its better scavenging effect, and is especially noticeable in high-speed engines. The amount of constriction in the area and the amount of taper of the port toward the outer part where the flange is attached, depends upon the speed of the engine.

In the Class B Military Truck Engine (page 49), both the inlet and the exhaust manifolds are on the same side of the cylinders side by side, as shown. Figure 112 shows a typical exhaust manifold for a four-cylinder engine; the two center branches are close together, for in four-cylinder engines, cast in pairs, the exhaust ports are always at the ends of each pair of cylinders and the inlet ports in the center. But even with cylinders cast en block, the exhaust ports of the center cylinders will as a rule discharge directly to the outside of the cylinders instead of in a common passage, between two adjacent cylinders. The exhaust

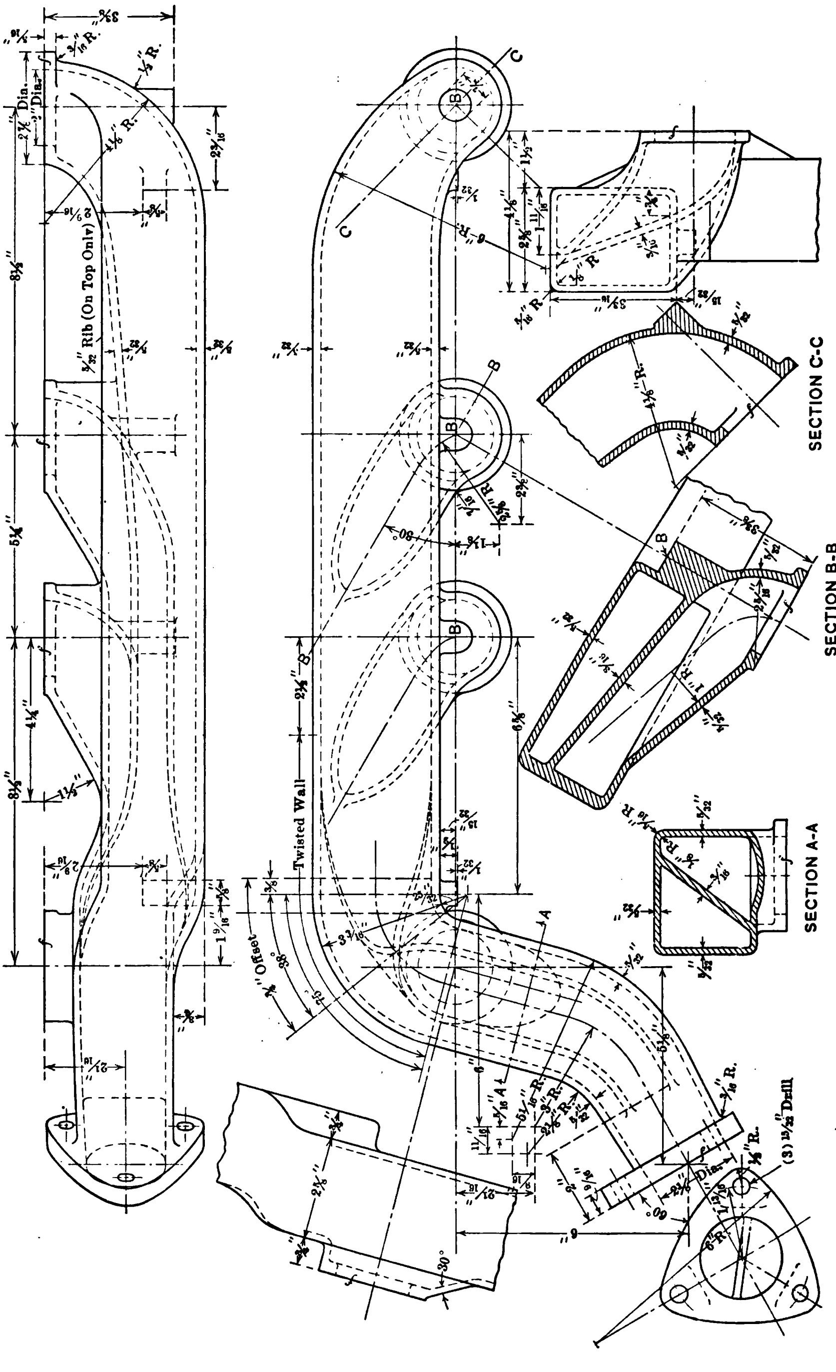


FIG. 113.—Exhaust manifold of Mack engine.

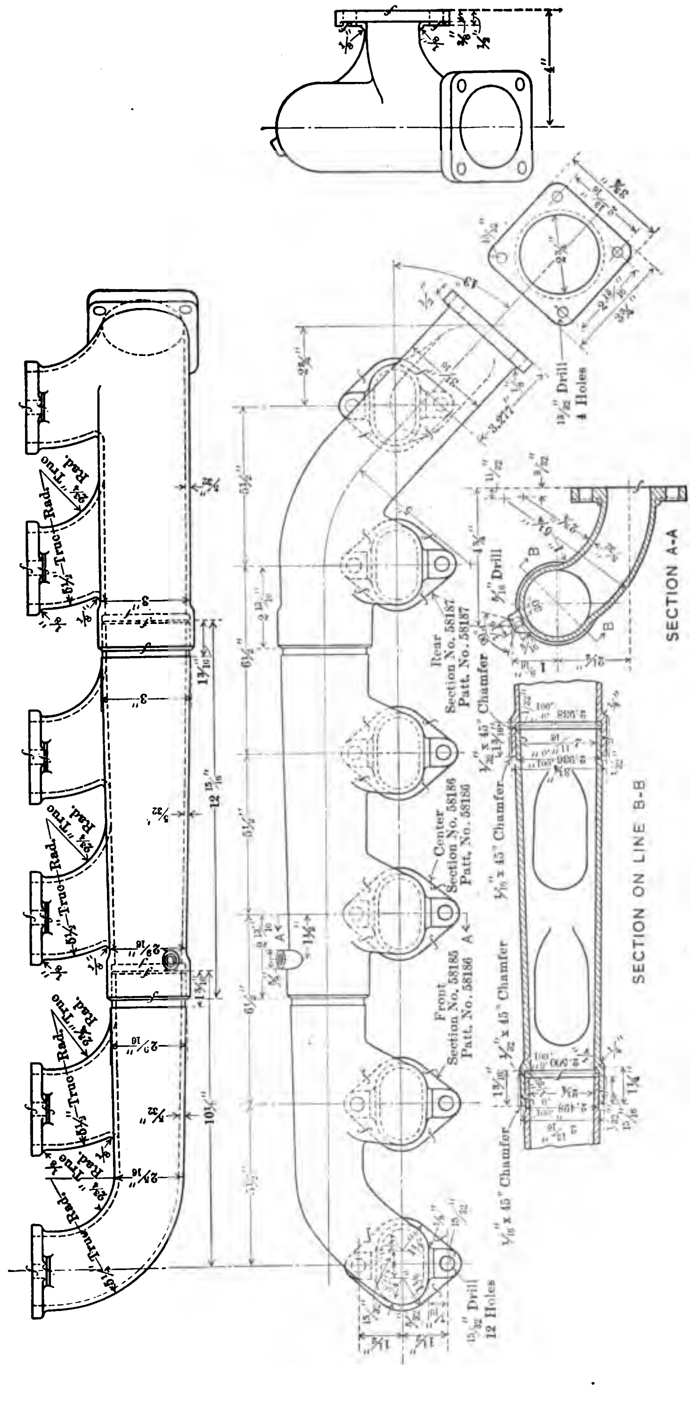


Fig. 114.—Exhaust manifold of twenty-four valve Pierce Arrow motor.

manifold may be made tapered, being gradually widened as more cylinders discharge into it. In a four-cylinder engine the exhaust manifold diameter is about one-half the cylinder bore at its widest section while in a six-cylinder engine it is about 60 per cent. of the bore.

Figure 113 shows a detailed drawing of the exhaust manifold of the Mack Engine referred to. Attention is drawn to the partition in the manifold itself, in order to prevent the flow of the exhaust gases from one cylinder into the other, when their exhaust valves may be open at the same time. The exhaust manifold is usually made of cast iron. The bosses *B* are the seats for the clamps by which the manifold is secured to the cylinders.

Figure 114 shows the exhaust manifold of the twenty-four valve Pierce Arrow engine made of malleable cast iron. The exhaust pipe tapers from a clear diameter of 2 inches at the front to 2¾ inches at the rear. The manifold is attached to the cylinder by flanges, each drilled with two $^{15}/_{32}$ inch holes for studs.

CHAPTER XVIII

ENGINE COOLING, COMBUSTION CHAMBER, ETC.

In all internal combustion engines, means must be provided for cooling the cylinder walls, either by surrounding the cylinder with a water jacket or by using the air direct as a cooling agent. If no provisions are made for cooling, the great heat developed within the combustion chamber, and, in addition, the friction between piston and cylinder, will cause the piston to bind or seize, decompose the lubricating oil, and cause premature explosions. On the other hand, the efficiency of all engines depends on the ability to turn into useful work the greatest possible number of heat units generated in the explosion chamber. It is evident, therefore, that the greater the number of heat units absorbed or lost by the cooling agent, the lower is the efficiency of the engine.

With very few exceptions all the internal combustion engines used on motor vehicles are water cooled, where the cooling system consists of the water jacket, the circulating system, the radiator, and the fan. It was shown in a number of illustrations that the water jacket is usually deeper around the cylinder head, as this part of the engine is exposed longest to the hot gases, and has to absorb and dissipate more heat. Directing the flow of the water coming from the radiator around the cylinders in a proper manner is also very important. In the design of the water jacket, ribs should be avoided as far as possible, as they will retard the flow of the water, and are liable to form steam pockets. In most designs the water will leave the cylinders at the highest point; this eliminates the formation of steam pockets.

Between the radiator and the cylinders the water-pump is interposed, which pumps the water from the bottom of the radiator into the water jacket. From here it flows from the jacket outlet into the radiator.

In the thermo-syphon system, the circulation is obtained by the fact that hot water will rise to the top while cool water will sink to the bottom. As the cylinder temperature increases the water will enter the water jacket at the bottom, rise to the top

of the cylinders by gravity, and thence flow into the radiator. To obtain good results it is necessary that there be provided plenty of water near the top of the radiator, so that the head pipe from the cylinders be submerged, and there should be an incline of the pipe from the cylinders to the top of the radiator.

It has been found from experiments with some engines that by increasing the temperature of the jacket water from 100°F. to 200°F. a saving of about 10 per cent. of fuel was obtained. Furthermore, the efficiency increases with the velocity of the expansion of the burning gases. By increasing the speed of the engine (that is to say, the piston speed) a higher fuel efficiency is obtained. The ignited gases in this case expand so rapidly (temperature and pressure fall more rapidly), and the time of contact between cylinder wall and the hot gases is so short, that a comparatively smaller number of heat units are absorbed or carried away by the walls of the combustion chamber and the cylinder at each explosion.

Furthermore, the combustion of the charge takes place gradually, and the greatest effective pressure is obtained from a charge highly compressed prior to ignition. In high-compression motors the clearance space is smaller; consequently, less cooling is required. The number of heat units lost through the cylinder wall would thus appear to be less in high-compression than in low-compression motors.

It is, of course, desirable to reduce the losses incurred by the cooling agent, unfortunately, most of the energy or the heat units derived from gasoline are lost in internal combustion engines, either through the exhaust or through the cylinder walls. Approximately 33 per cent. of the heat energy is lost through the wall.

ENERGY AND HEAT ENERGY

The mechanical energy in a motor is measured by horsepower or foot pounds.

One foot pound is the energy required or expended in raising a weight of 1 lb. to a height of 1 ft. Heat energy on the other hand is measured in British Thermal Units (B.t.u.).

One B.t.u. is the amount of heat required or expended in raising the temperature of 1 lb. of water one degree, from 63° to 64°F.

One B.t.u. is equal to 778 ft.-lb.

One pound of gasoline upon combustion will liberate about 19,000 B.t.u.'s.

FUEL CONSUMPTION

In testing an engine for fuel consumption it is found that the amount of fuel used per horsepower per hour, varies with the amount of power furnished. When delivering a small amount of their maximum power, engines are usually wasteful of fuel; the fuel efficiency gradually rising with an increase in power to about ¾ full load, after which it decreases again. Under favorable conditions, at maximum horsepower, engines consume about .65 lb. of gasoline per horsepower per hour. To find the amount of gasoline required per hour for a given engine, multiply the horsepower by .65. For instance, if the engine horsepower is 50, the gasoline consumed per hour would be: $50 \times .65 = 32.5$ lb. If we assume that 33 per cent. of the heat energy is absorbed by the jacket, in a 50-horsepower motor, the jacket loss would be: $32.5 \times 19{,}000 \times .33 = 203{,}775$ B.t.u.'s, or $\frac{203{,}775}{60} = 3396$ B.t.u.'s per minute.

Ordinarily, the engine will consume a great deal more than .65 lb. of gasoline per horsepower per hour; on the other hand it is very seldom that the engine is run at its full capacity for a considerable length of time, except in tractors.

DIMENSIONS OF WATER PIPES

In figuring the size of the water inlet and outlet pipes, there is quite a difference in the corresponding sizes of engines on the market. For pump circulation 3500 B.t.u.'s per minute, may be allowed per square inch of pipe area, for touring cars, and 3000 B.t.u.'s for truck motors. For thermo-syphon cooling 1000 B.t.u.'s per square inch may be allowed. In the above example 3396 B.t.u.'s were lost to the cooling agent per minute. The outlet pipe would therefore have an area of $\frac{3396}{3000} = 1.13$ square inches, for a truck motor.

Frequently the inlet is made somewhat smaller than the outlet, the difference usually being ⅛ inch to ¼ inch in the diameter. The size of the radiator and the fan will also affect the amount of water necessary, for if the water enters the jacket at a lower temperature it will be a more efficient cooling agent than if the radiator and the fan are less capable of lowering its temperature.

At times better cooling is obtained when the water jacket is not too large, so that the quantity of water in the jacket is more rapidly changed than if the jacket capacity were larger. In the last case, the water would flow past the cylinder walls at a slower speed and leave the cylinder outlet at a higher temperature.

In the beginning of this chapter it was shown that in order to obtain the highest efficiency the cylinder should be cooled neither too much nor too little. With the ordinary system of pump circulation the cooling is in a measure proportional to the engine and car speed, and it is also dependent on the weather temperature. If the car is running at fairly high speed on a level road with the throttle say, about one-half open, the pump, running at high speed, will force a large quantity of water through the water jacket, and the radiator would cool the water very efficiently. In such a case the engine may be over-cooled and for this reason the fuel efficiency will not be satisfactory. On the other hand, when climbing a hill with the throttle wide open the engine may become over-heated.

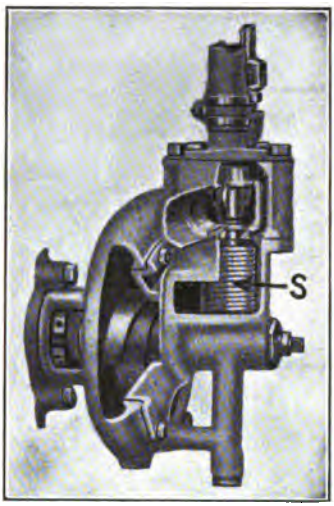


FIG. 115.—Sylphon thermostat of Cadillac motor.

It is evident, that if the water could be kept at a constant temperature—a temperature at which the fuel efficiency is highest—without causing pre-ignition in the engine or excessive friction between piston and cylinder wall, more "miles" per gallon of gasoline would be the result.

To obtain this effect some manufacturers use a thermostatically controlled water circulation. In this system a uniform temperature of the water is obtained by valves opened and closed by a sylphon thermostat. This device itself is a bellows-like container filled with a liquid, which, when heated is transformed into a gas. The resulting pressure elongates the thermostat, forcing a valve to close or partly close the water flow.

In the Cadillac eight-cylinder motor the thermostat is attached to the pump (see Fig. 115) where S is the sylphon.

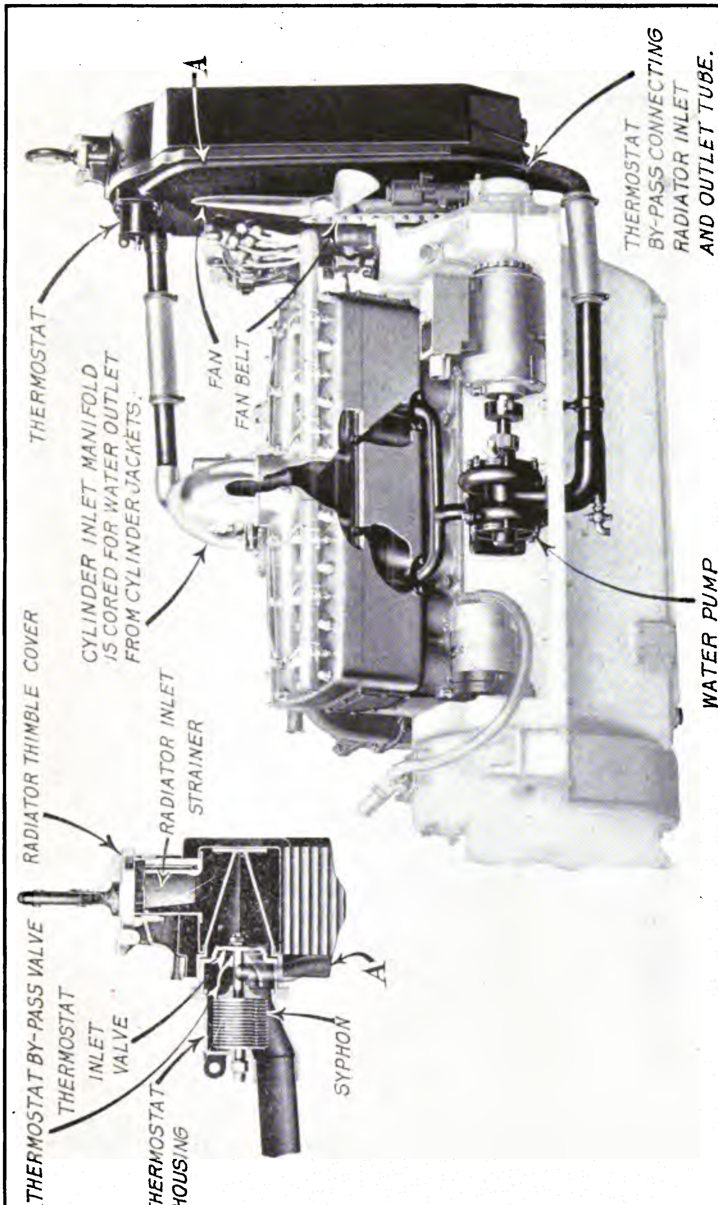


Fig. 116.—Syphon thermostat and cooling system of Packard Motor.

Connections to the thermostat housings allow the pump to draw water from the radiator, the cylinder blocks, and the water jacket of the intake manifold.

When the engine is cold, the thermostat valves are closed, and the supply from the radiator is cut off. The water then is

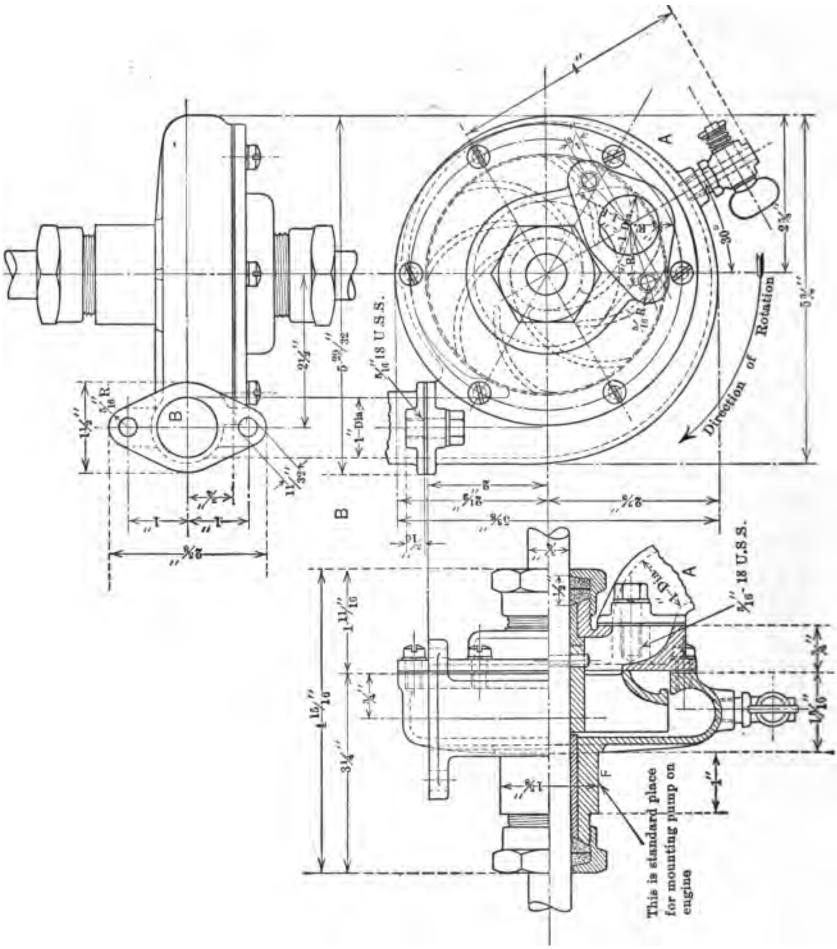


FIG. 117.—Fulflo water-pump.

circulated only through the water jackets of the cylinder heads and intake manifold. Because of the fact that the amount of water to be heated is relatively small, the temperature of the cylinders and intake manifold quickly rises to the point at which the engine operates best.

As soon as the temperature of the water in the water jackets reaches the predetermined point, the thermostat forces open the valve and allows only sufficient cooled water from the radiator to enter to keep the temperature down to a predetermined point.

In the Packard trucks the thermostat is located on the top of the cylinder. In the Packard twin-six passenger car it is attached to the upper tank of the radiator (Fig. 116), and by-passes the water to the inlet side of the pump through pipe *A* until the motor has reached the proper temperature for efficient running. This illustration shows the water from the pump admitted to the cylinder casting at divided points at the bottom of the water jacket.

WATER PUMP

Figure 117 shows details of the Fulflo pump for motor vehicle water circulating systems which is giving satisfaction. This pump, when run at 1750 to 1800 r.p.m. circulates approximately 20 gal. of water per minute. For smaller motors where less water is required, the speed may be reduced to as low as 1000 revolutions per minute with a corresponding reduction in the amount of water forced through the jacket. Instead of reducing the speed some manufacturers simply use reducers on the outlet connections, making the outlet from one-half to three-fourths as desired. The ideal would of course be to run the pump at its normal speed and to use the thermostat for controlling the amount of water force through the engine water jacket.

The water coming from the radiator enters the pump at *A* and leaves it at *B*. The water entering the pump near the center is forced out by the centrifugal action set up by the blades shown in the drawing.

CHAPTER XIX

POWER OF ENGINES; EFFICIENCY

The power of an engine is expressed in horsepower (H.P.). One H.P. = 33,000 ft.-lb. per minute or $\frac{33.000}{60} = 550$ ft. lb. per second.

Expressed in Heat Units (British Thermal Units) One H.P. = $\frac{33,000}{778} = 42.416$ heat units per minute (Often written 42.416 B. T. U.) since One B. T. U. = 778 ft.-lb.

The fundamental equation for the horsepower (H.P.) of any prime mover, *i.e.*, any engine which develops power from fuel, like a steam engine, is

$$\text{HP} = \frac{PLAM}{33,000}$$

where P = the mean effective pressure in pounds per square inch on the piston head during the power stroke (see Chapter XXI).

L = the length of the piston stroke in feet.

A = the area of the piston head in square inches.

M = the number of power strokes per minute.

In a four-cycle internal combustion engine the pressure does not occur at every stroke of the piston, as in most steam engines, but only once in every four strokes or once in every two revolutions of the crank shaft. Therefore we may write the horsepower (H.P.) of a four-cycle engine is $\text{H.P.} = \frac{PLARN}{33,000 \times 2}$ where N = the number of cylinders and R the number of revolutions per minute. If instead of the length of piston stroke we have the piston speed in feet per minute S ($S = 2LR$), by substituting S for the piston stroke and r.p.m., the formula is,

$$\text{H.P.} = \frac{PASN}{33,000 \times 4}.$$

The HP so found, is called the theoretical HP or the indicated horsepower (IHP).

The *Mean Effective Pressure P* (see Chapter XXI) is the average pressure acting on the piston head during the entire expansion stroke, less the average pressure during the entire compression and exhaust strokes. The last named is very small in internal combustion engines, and may be neglected ordinarily.

The Brake Horse Power (BHP) of an engine, is the power actually delivered by the engine crankshaft; it is the power available for doing useful work. It is approximately equal to the IHP minus the friction losses F in the engine. Thus BHP $=$ IHP $-F$, and $F =$ I.H.P. $-$ B.H.P.

MECHANICAL EFFICIENCY

The mechanical efficiency of an engine is the BHP divided by the IHP. If E, is the mechanical efficiency, $E = \frac{\text{BHP}}{\text{IHP}}$ and BHP $=$ IHP $\times E$. Using the former equation, the Brake Horse Power $= \frac{PASNE}{4 \times 33{,}000}$, where $N =$ number of cylinders. The mechanical efficiency is not constant but varies with the load and the speed.

Thermal Efficiency.—The thermal efficiency of an internal combustion engine is the ratio of the BHP to the power in heat units contained in the fuel consumed.

$$H = \frac{\text{BHP (expressed in heat units)}}{\text{Heat units in the fuel consumed.}}$$

When speaking of Heat Energy, we stated that one B.t.u. $=$ 778 ft. lb. We have also seen that one H.P. $=$ 33,000 ft. lb., per minute. Therefore one H.P. per minute $= \frac{33{,}000}{778} = 42.42$ B.t.u.'s, and one H.P. per hour requires

$$\frac{60 \times 3{,}3000}{778} = 2545 \text{ B.t.u.'s.}$$

If there were no losses in an engine, one B.t.u. per minute would give 778 ft. lb. per minute behind the piston, and 2545 B.t.u.'s would develop one H.P. per hour.

However, the number of heat units lost in an engine is very large; only about 20 per cent. being utilized or given off by the

engine in actual power. Fuel oil or gasoline as in use today, contains about 19,000 B.t.u.'s per lb. of fuel. With these figures we can calculate the amount of power we should obtain from 1 lb. of gasoline if there were no losses, and can find the thermal efficiency H.

For example, if .65 lb. of gasoline is used per B.H.P. per hour in a certain engine, or $\frac{19{,}000 \times .65}{60} = 205.8$ B.t.u.'s per minute, and one HP as we have seen = 42.42 B.t.u.'s per minute, then in this engine the thermal efficiency $H = \frac{42.42}{205.8} =$ approx. .206 or 20.6 per cent.

The formula mostly used by manufacturers for determining the HP of their engines (as advertised in their catalogue), which is usually called the N.A.C.C. (National Automobile Chamber of Commerce) Rating, and which is also used by the Royal Automobile Club of Great Britian, is HP $= \frac{D^2N}{2.5}$; in which D is the diameter of the cylinder in inches. In the N. A. C. C. Rating, it is assumed that the motor delivers its rated power at a piston speed of 1000 ft., per minute, and that the mean effective pressure P, in the cylinders will average 90 lb. per square inch; it is further assumed that the mechanical efficiency E, will average 75 per cent. By substituting these values in the above equation and using the formula, Area $= .7854 \times D,^2$ we have the complete equation:

$$\text{B. H. P.} = \frac{90 \times .7854 \times D^2 \times 1000 \times N \times .75}{4 \times 33{,}000}$$

Working out the figures, we have B.H.P. $= \frac{D^2N}{2.489}$ or approximately, $\frac{D^2N}{2.5}$.

If we call L the length of the piston stroke in feet, then the work done in one cylinder during one cycle of operation is

$$.7854 \times D^2 \times P \times L = \text{foot pounds.}$$

If the piston stroke is given in inches we simply divide the equation by 12. If the number of revolutions of the motor per minute is called R, then the number of explosions per minute will be $\frac{R}{2}$ (in a four-cycle engine), and the power developed per minute in a motor with N cylinders is $\frac{.7854\ D^2RPLN}{2 \times 33{,}000} =$ I.H.P.

The indicated horse power, is so called as it is the power theoretically conveyed by the expanding gas to the moving piston. A certain amount of power is lost due to friction of the piston and the bearings, the valve operation, etc. If the mechanical efficiency of the motor is designated by E, our formula for the actual power delivered by the motor becomes

$$\text{B. H. P.} = \frac{.7854\ D^2RPLNE}{2 \times 33{,}000}$$

If S is the piston speed in feet per minute, $S = 2\ LR$. Substituting this in the foregoing equation for brake horsepower, we have

$$\text{B.H.P.} = \frac{.7854D^2\ SPNE}{2 \times 33{,}000 \times 2} = \frac{D^2\ SPNE}{168{,}000\ (\text{approx.})}.$$

Let us assume that the mean effective pressure P, when the engine is giving its maximum power, is 90 lb. and that it gives this power at a piston speed S, of 1500 feet per minute; let us further assume that the mechanical efficiency is 75 per cent. With a four-cylinder engine and a bore of 5 inches, our formula becomes

$$\text{B.H.P.} = \frac{5^2 \times 1500 \times 90 \times 4 \times .75}{168{,}000} = 60 \text{ approx.}$$

CHAPTER XX

BRAKE-HORSEPOWER TESTS OF AUTOMOBILE ENGINES

In order to find the actual H.P. of an engine delivered by the crankshaft, it is necessary to make a test with a dynamometer, either an absorption dynamometer which absorbs the energy given out by a pulley attached to the crankshaft, or by a transmission dynamometer which is inserted between the engine crankshaft and the load, and in which the dynamometer in itself does not absorb any of the power. Almost all the dynamometers in use to-day are absorption dynamometers. It is customary to express the power of an engine in horsepower, taking 33,000 ft.-lb. per minute as equal to one horsepower. In other, words, the power necessary to raise 33,000 lb., to the height of 1 foot in 1 minute, *i.e.*, 33,000 ft.-lb. of work per minute, is equal to 1 H.P. (The term horsepower is not accurate, for it has been found that the average horse working 8 hours per day, works only a little over two-thirds of the theoretical horsepower mentioned above.)

In practice it is found that the actual H.P. which is delivered by the crankshaft, called the Brake Horsepower (B.H.P.) is about 15 per cent. to 25 per cent. less than the Indicated Horsepower (I.H.P.) in well-designed, up-to-date motors; in many engines it is a great deal less. The difference between the I.H.P. and the B.H.P. is the loss due to friction in the engine itself.

The B.H.P. then, is the actual power of the engine which is available. One of the simplest ways to measure this power is on the flywheel or on a pulley, keyed to the crankshaft of the engine, which is made to run against a certain resistance which can be calculated.

The underlying principle of the most common device used for determining the B.H.P. of an engine is shown in Fig. 118. F is a pulley or the flywheel of the engine, W_1 and W_2 are two weights which are exactly balanced when the engine is running in the direction indicated by the arrow, and b is a belt. d is the radius of

the pulley plus one-half the thickness of the belt, expressed in feet; R, the number of revolutions per minute. Then the B.H.P. of the engine $= \frac{2\pi dR(W_1 - W_2)}{33{,}000}$. Dividing 2π by 33,000, that is to say $\frac{2 \times 3.14159}{33{,}000} = \frac{1}{5252.1} = .0001904$; our formula then becomes B.H.P. $= .0001904dR\,(W_1 - W_2) = \frac{dR(W_1 - W_2)}{5252.1}$. In practice it is not found convenient to balance the two weights when changing the speed, and for this reason, the Prony brake shown in Fig. 119 is found more convenient, while for larger engines, that shown in Fig. 120, or a similar device, is often used.

The formula for finding the H.P. with the brake shown in Fig. 119 is, B.H.P. $= .0001904dR$ (W + weight of spring balance), where d is the distance in feet as shown, R, the number of rev-

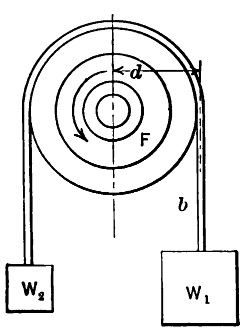


FIG. 118.—Simple brake.

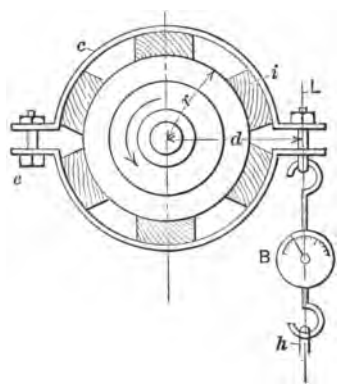


FIG. 119.—Prony brake.

olutions per minute (r.p.m.) and W, the pull indicated on the spring balance in pounds. This simple form of Prony brake is composed of a metal band c, to which are attached a number of wooden blocks i. As is seen, the brake band is made of two pieces having bolts on both sides, to be able to apply pressure between the brake and the pulley.

B is the spring balance, the lower end of which is fastened to a hook h on the floor. The brake should be so constructed that both sides of it will balance each other when resting on the pulley. The spring balance should then be weighed as the weight of it must be added to W when figuring the H.P.

When the engine is running the mixture is adjusted in order to

obtain the best pull on the spring balance. The bolts are tightened for a certain number of revolutions per minute, or a certain pull, as desired. When the engine is running smoothly and every thing is properly adjusted, find the number of revolutions of the crankshaft per minute, (by means of a speed indicator or speed counter), and at the same time note the pull indicated on the spring balance. Suppose for instance that the r.p.m. is 1000 and the pull on the balance indicates 35 lb., and if the distance between the center of the shaft and the center line L through the spring balance is 15 inches, *i.e.*, 1.25 feet (d must be measured at right angles to line L), then the formula for finding the Brake Horsepower

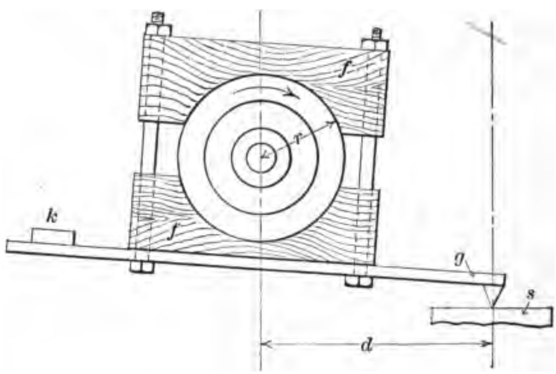


FIG. 120.—Prony brake for testing power of engines.

B.H.P. = $.0001904 \times 1.25 \times 1000 \times (35 +$ the weight of the spring balance.)
Supposing the weight of the spring balance to be 5 lb.,

B.H.P. = $.0001904 \times 1.25 \times 1000 \times (35 + 5) = 9.52$ horsepower. Another form of Prony brake which is found convenient for larger engines is that shown in Fig. 120. In this case, ff are 2 wooden blocks, g is an iron bar, S a platform scale, and K a balance weight, to balance the apparatus on the pulley. When using this type of Prony brake the formula is

$$\text{B.H.P.} = .0001904dWR.$$

For example, if we test an automobile engine and get the following readings: Number of revolutions per minute 1100, reading on scale 75 lb., d, the horizontal distance between center of pulley and the center line of contact between g and S is 3 feet. Then the B.H.P. = $.0001904 \times 3 \times 1100 \times 75 = 47.1$.

If the weight of the arm, g is not balanced by a balance weight K on the opposite side, then the weight of the arm bearing down on the scale should be deducted from W in the equation. By looking at Fig. 119 or 120, it can be seen that the friction or the pull on the circumference of the pulley depends on the ratio of d to r, and on W, it is $\frac{d}{r} \times W$ (r = radius of pulley).

The velocity or the speed of the face of the pulley, *i.e.*, the rubbing speed on the pulley is $2\pi rR$, from which we may write

$$\text{B.H.P.} = \frac{\frac{d}{r} \times W \times 2\pi r \times R}{33{,}000}.$$

By canceling the r's in the numerator of the equation, we have

$$\text{B.H.P.} = \frac{2\pi dR \times W}{33{,}000},$$

and as we have seen before, $\frac{2\pi}{33{,}000} = \frac{1}{5252.1} = .0001904.$

Therefore we may also write $\text{B.H.P.} = \frac{dRW}{5252.1}$

TORQUE

The force which tends to turn or twist the crankshaft (*i.e.*, the turning effort or the twisting moment) is clearly the distance d multiplied by W, and this is called the torque T.

$T = dW$, and is expressed in pound feet; or in pound inches, if d is given in inches. This value dW is in a sense independent of the power of the motor, for T may be low, yet if R is high the H.P. will be high.

The above formula may therefore also be written,

$$\text{B.H.P.} = \frac{TR}{5252.1},$$

and if it is desired to find T when only the H.P. and R are known,

$$T = \frac{\text{B.H.P.} \times 5252.1}{R} \text{ pound feet.}$$

If instead of the B.H.P. we only know the I.H.P. and wish to

find T on the I.H.P., we simply substitute I.H.P. for B.H.P. in this formula. Thus, the torque on the I.H.P. is

$$T = \frac{\text{I.H.P.} \times 5252.1}{R} \text{ pound feet, or}$$

$$T = \frac{\text{I.H.P.} \times 5252.1 \times 12}{R} \text{ pound inches.}$$

When testing the B.H.P. of an engine a number of readings should be taken at different speeds, which is done by tightening the brake more or less. Find a number of different values of the

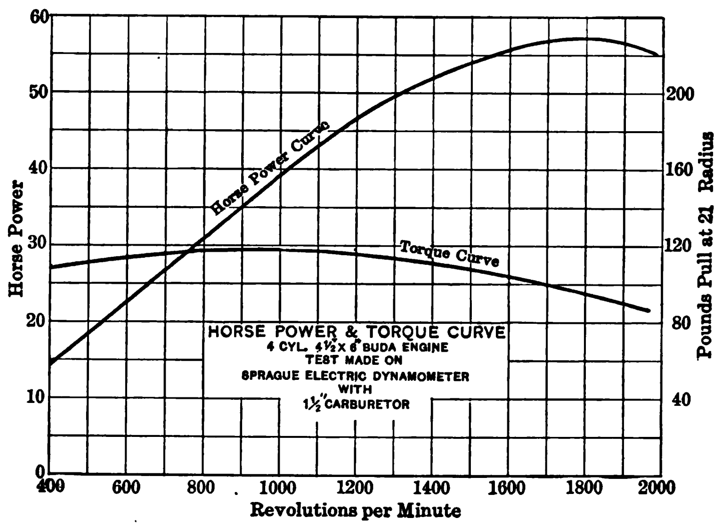


FIG. 121.

B.H.P. for different speeds of the engine, and see how the H.P. increases up to a certain speed when it begins to decline. (It is interesting to find the H.P. given out by the engine at various speeds, as the automobile does not always run at its maximum speed.) The result of the test is then plotted, in curves, as shown on Fig. 121.

When the maximum B.H.P. of an engine has been found the engine should be tested for consumption of gasoline per B.H.P. per hour. In order to do this the gasoline used for running the engine must be weighed or measured, and the engine be allowed to run at least 15 minutes at the maximum load.

Tests should always be taken for various loads to see the dif-

ference in consumption of gasoline per B.H.P. when running on full load and on smaller loads and to find the load at which the gasoline consumption is the lowest. It might be mentioned in making brake tests, the engine should be permitted to run for some time, in order to be warmed up, before the actual test is made; the effect of a cold cylinder wall upon the efficiency was shown in former chapters.

With all forms of Prony brakes, means should be provided for cooling the pulley as the heat generated by the friction is enormous. This is usually accomplished by using a pulley having a rim with inwardly turned flanges and plying a stream of water into the pulley. There are other forms of absorption brakes, as for instance, the rope brake, but the principle of their operation is the same as those described.

CHAPTER XXI

MEAN EFFECTIVE PRESSURE

The Mean Effective Pressure P, can be calculated when the torque of the motor is known and its volume. P is usually expressed in pounds per square inch.

In Chapter XX it was shown that the

$$\text{B.H.P.} = \frac{2\pi RdW}{33{,}000} = \frac{2\pi RT}{33{,}000},$$

where T is the torque and $= dW$.

In Chapter XIX we have also seen that the H.P. $= \dfrac{PLARN}{2 \times 33{,}000}$.

A being the area of the piston in square inches, R the revolutions per minute, N the number of cylinders and L the piston stroke in feet. If L is given in inches, we have

$$\text{H.P.} = \frac{PLARN}{2 \times 33{,}000 \times 12}.$$

The volume V of the cylinders in cubic inches swept by the pistons is LAN; substituting V in the last equation and solving it for P, we have

$$P = \frac{\text{H.P.} \times 2 \times 33{,}000 \times 12}{VR}.$$

If we here substitute for H.P. the value above found for the brake Horsepower, we have

$$P = \frac{2\pi RT}{33{,}000} \times \frac{2 \times 33{,}000 \times 12}{VR} = \frac{150.8T}{V},$$

V being the volume of all the cylinders.

The value for P found in this manner from the Brake Horsepower is the Brake Mean Effective Pressure.

The Mean Effective Pressure on the I.H.P. can be found from the equation for H.P. given in Chapter XIX.

$$\text{H.P.} = \frac{PLARN}{2 \times 33{,}000}, \text{ whence } P = \frac{\text{H.P.} \times 2 \times 33{,}000}{LARN}$$

where L is expressed in feet, and A in square inches; N is the number of cylinders.

When the indicator diagram of an engine is known the mean effective pressure may be determined graphically as follows:

Figure 122 represents an indicator diagram with the exhaust and suction curves rather exaggerated. In practice, usually only the compression curve is deducted from the height of the expansion curve; except when the exhaust or suction lines are very large, they too have to be deducted, as they represent negative work. The length of the diagram is divided into ten equal spaces as shown by dotted lines. In the center of each space

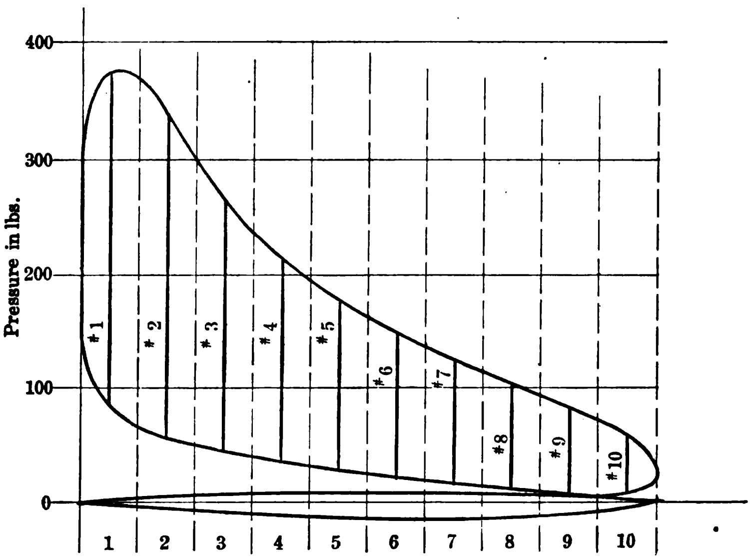


FIG. 122.—Diagram for finding mean effective pressure.

a solid line is drawn, the length of which indicates the mean height, and thus the mean pressure on the diagram for such space. The lengths of these lines numbered 1 to 10 will therefore represent the mean (average) pressures in the spaces in which they are located. All the lengths of these ten lines are then measured and added, and the result is divided by the number of lines, in this case 10; in other words the mean effective pressure.

$$P = \frac{\#1 + \#2 + \ldots\ldots \#10}{10}$$

The scale of the vertical distances must be selected that each

100 lb. of pressure represent unity. For instance, if 100 lb. in height measures 1 inch, then the mean pressure can be scaled off directly in decimals of an inch, and the result multiplied by 100.

The indicator diagram may be determined with a manograph which is described in Chapter XXII.

The greater the number of equal spaces into which the horizontal distance of the diagram is divided, the greater will be the accuracy. Naturally, if it were divided into 20, the mean heights of the 20 spaces must be added and the result divided by 20.

CHAPTER XXII

THE MANOGRAPH[1]

The Manograph, reference to which has been frequently made, is an instrument which indicates the pressure inside the cylinder at the different piston positions. Its operation is as follows (see Fig. 123):

A small mirror is pivoted in such a way as to move about a horizontal axis under the influence of the pressure exerted upon a diaphragm.

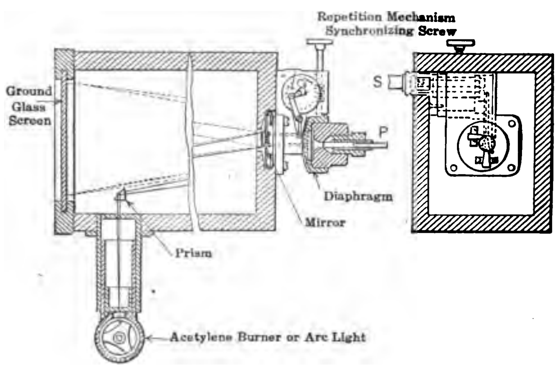


FIG. 123.—The manograph.

The diaphragm—represented by a heavy black line is held in place by a screw gland which enables a change of diaphragm to be effected readily.

A second motion is given to this mirror, causing it to move about a vertical axis in synchronism with the piston of the engine.

Each of the two movements of the mirror taken separately would result in a straight line being shown by the beam of light on the ground-glass screen. When pressure varies in the cylinder and the piston moves simultaneously, the beam of light moves over the screen in accordance with the value of the two components and traces a line which forms a diagram. That movement

[1] See Hugo C. Gibson in the S. A. E. Transactions, Vol. 8, Part I, from which the illustration and a large part of the text were taken.

of the mirror which corresponds with the piston movement is effected by the flexible shaft which is attached to the crankshaft of the engine or similar rotating part, the rotary motion thus obtained being transformed into the above-described motion of the mirror by means of a small crank. The pressure movement of the mirror is produced by means of a diaphragm, which is connected with the cylinder by means of a small pipe, *P*.

Diagrams showing the low pressures of suction, or the high pressures of combustion, may be obtained with the same instrument, it being only necessary to change the diaphragm, the same as the spring would be changed in an ordinary indicator.

The method of attachment to an engine is simple; the flexible shaft *S* being provided with a nose which is screwed into a tapped hole in the end of the crankshaft, and the pressure pipe being ordinarily provided with a steel cock to be screwed into the compression-cock hole, a valve cap, or other convenient position.

For visual observation the tracing of the diagram by the beam of light is observed on the ground-glass screen. Temporary records may be made by following the line with an ordinary pencil. Alteration of engine adjustment or other conditions may be noted, and the result also traced with the pencil and the difference recorded; thus no effort of memory is necessary to compare diagrams.

Permanent records are made by substituting a photographic dark slide for the ground-glass screen, and exposing a plate of film to the action of the beam light; the negative is developed and printed from it in the ordinary way, thus securing a record from which data may be taken for the determination of mean effective pressures, indicated horsepower, etc.

This instrument is so sensitive that it will indicate small waves on the combustion stroke which it is claimed are produced by pulsations or surging waves in the pressure pipe. These small waves can be eliminated by means of coiled wires inserted in the pressure pipe as suggested by Dr. Watson.

The instrument being calibrated, the pressure in pounds per square inch can be read by the light of the indicator diagram.

When the manograph is connected to the engine it must be adjusted for phase equality. This is done by means of the synchronizing thumb screw. This screw is turned to the right or the left, until, when the motor is turned over by hand, the pressure curves on the compression and expansion strokes follow

practically the same line. If the expansion line does not follow very closely the compression line, but lies more than a small amount below, during the first part of the expansion stroke (when the motor is turned by hand), it indicates excessive cylinder leakage.

When the engine is operating normally, the pressure on explosion will be about four times that of compression, depending upon the quality of the explosive mixture.

The manograph may also be used for the rapidly varying pressures in the intake and exhaust pipes.

When using the weak-spring in the instrument the compression pressure may be determined, also if the exhaust from one cylinder enters the other, as is frequently the case in six-cylinder engines; this defect can easily be overcome by dividing the exhaust manifold as shown in Fig. 113, so that no two cylinders, whose exhaust valves are open at the same time, discharge in close proximity into the same compartment.

In the application of a manograph to a multi-cylinder engine, it is advisable to use one instrument only for all cylinders (on one cylinder at one time, of course), and not a separate manograph for each cylinder, because it is almost impossible to so tune each manograph as to give cards of identical characteristics. We must recognize that each diagram and mirror mechanism has its own calibration scale and will give a correct diagram in accordance therewith, but that no two of these combinations will give exactly similar cards under exactly similar influences. If separate manographs were applied to the same cylinder each card would vary in accordance with the calibration of the instrument producing it, although each would be true in the light of its own calibration. With a separate instrument for each cylinder the variation in cards viewed collectively is confusing.

CHAPTER XXIII

BRAKE-HORSEPOWER TEST BY ELECTRIC DYNAMOMETER[1]

The electric dynamometer is a machine much used for testing high-speed motor vehicle and airplane engines. It is primarily an absorption brake for measuring the horsepower delivered by an engine, but has many other applications, and is equally able to measure the torque delivered, which with the speed gives the Brake Horsepower. It affords the simplest means to measure the friction losses, and thus the mechanical efficiency of a motor.

HORSEPOWER IS INDICATED

Horsepower is the product of torque (pounds-feet) and speed (r.p.m.) multiplied by a constant. The electric dynamometer measures the horsepower of an engine by indicating on a scales the torque delivered by the engine shaft, while a tachometer connected to the shaft indicates the speed. In a size of dynamometer much used the constant equals $\frac{1}{3000}$. With such a machine the horsepower output of the engine in test equals the pull in pounds indicated on the scales attached to the dynamometer, multiplied by the speed in r.p.m. as indicated on the tachometer, divided by 3000.

The constant of 3000 is arrived at, in the following way: The regular Prony Brake Formula applies:

$$\text{H.P.} = \frac{2 \times \pi \times R/12 \times \text{Lb.} \times \text{R.P.M.}}{33{,}000}.$$

In this formula R represents the length of torque arm. If this torque arm is 21.008″, the formula reduces to

$$\text{H.P.} = \frac{\text{Lb.} \times \text{R.P.M.},}{3000}, \quad \text{since} \frac{2\pi \times \frac{21.008}{12}}{33{,}000} = \frac{1}{3000}.$$

[1] This chapter to the end of the table, page 233, has been prepared by Mr. Carl F. Scott, of the Sprague Electric Works, New York.

In the same manner if the torque arm is 15¾″ the constant is 4000 and if it is 63.025″ it is 1000.

WHY HORSEPOWER TESTS ARE NEEDED

Horsepower tests are useful to the study of nearly every part of an engine. The value of a carburetor is shown by the results it produces in the way of power per unit of fuel consumed and the power it can effect in the engine at different speeds. The efficiency of an ignition system can be determined by power tests at varying speeds and loads. Power tests will also show the efficacy of intake manifold design, the effect of exhaust manifold and muffler, the results of elimination of vibration by balancing, and the difference in pistons and piston-rings.

Of great importance is the test of lubricating oils by studying their relative value under known conditions of load and speed. Water-pumps, oil-pumps, radiator fans, starting motors, lighting generators, transmission and bevel drives, tires, complete chassis, propellers for airplanes, can all be tested with the electric dynamometer.

DESCRIPTION OF DYNAMOMETER

The electric cradle dynamometer consists of an inner revolving rotor or armature which is coupled to the shaft of the engine in test, and an external field frame, or field structure, which supports the armature shaft by bearings mounted in brackets on either end of the frame. These brackets, which carry the armature bearings, and between which the frame is bolted, are provided with bosses or hubs, which are in turn supported in ball-bearing trunnions or pedestals. Supported in this way, the whole frame structure would be free to revolve around an axis concentric with the axis of the revolving armature, and would so revolve were it not restrained by a set of scales.

The frame carries the field poles, which are in fact electro-magnets, and which must be separately excited from a suitable 115 or 230 D.-C. supply circuit. The electro-magnetic reaction between the armature and field causes the latter to tend to rotate with a torque equal to that exerted by the armature. Furthermore, the small amount of friction of the armature bearings and the current-collecting brushes on the armature acts to pull on the field in the same direction. So it is evident that all of the

torque exerted by the engine shaft appears on the scales, no corrections being required.

HOW THE POWER OF THE ENGINE IS ABSORBED

The mechanical power of the engine appears in the dynamometer armature to which it is coupled. This mechanical power is transformed into electric energy in the form of electric current, which flows out of the dynamometer through the cables connected to the frame and into a resistance-box, where the energy is finally dissipated in heat. In some cases, however, the electric energy is not wasted as heat in a rheostat, but is used just as the current of a generator in a power plant is used, for electric lamps or electric motors.

HOW THE LOAD IS VARIED

In an electric dynamometer the revolution of the armature in the magnetic field or magnetism produced by the field coils on the frame results in currents being set up in the armature which are led outside the machine. These currents in turn react on the magnetism of the field and cause a counter-torque or restraining effort. The amount of this counter-torque, and therefore the amount of load placed on the engine, depends on two factors—the strength of the field, and the volume or magnitude of the current. The strength of the field is changed by turning the handle of the field rheostat on the switchboard, and this is one way, and the best way up to a certain point, of varying the load to be placed on an engine by the dynamometer. The volume of current can be varied by this same means, but it can also be increased by decreasing the resistance of the grids in which the current is consumed. Therefore the load can also be changed by changing the external resistance. This is done by the dial-switch on the control-panel, which varies the connections of the resistance. This external resistance is also used for starting the dynamometer, so that it can act as a motor to crank the engine or run the engine for making tests of the friction loss.

HOW A TEST IS RUN ON THE DYNAMOMETER

The engine is carefully set up, accurately aligned, coupled up, and gasoline and water connections made and all such preparations attended to. When all ready, the operator at the control-panel sees that the switches are in proper position, and then

pushes the starting button. When the engine begins to "fire," the man at the engine should "throttle down" and the man at the control-panel will throw over the load-switch so as to connect it from the supply line whence it derives its energy for starting. The control-panel operator then sets the load-switch to the point which he judges, or has found by trial, best suited to the engine in test. He then slowly strengthens the field of the dynamometer by turning the field rheostat until the load is increased to the

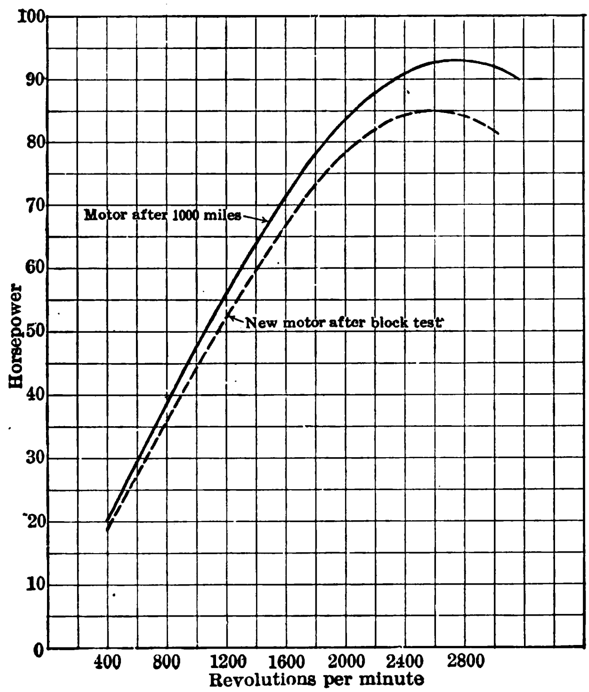


FIG. 124.—Power curves of the Packard twin-six motor.

desired point, the engine throttle being opened so that the engine will pull its full load.

MAKING A POWER CURVE

The two curves shown on the accompanying illustration (Fig. 124) were made in the following way:

The measurements were all made with wide open throttle, and the curves represent at any speed the maximum power the engine could exert at that speed.

The engine had been started by using the dynamometer as a motor, and it had then run "light" for a time, warming up a little, with the dynamometer coupled up, of course, but with no load on the dynamometer. That is, the field rheostat on the dynamometer panel was turned so as to put so much resistance in the field circuit that its magnetism was reduced and the armature would not have currents set up in it, even if the load-switch were closed.

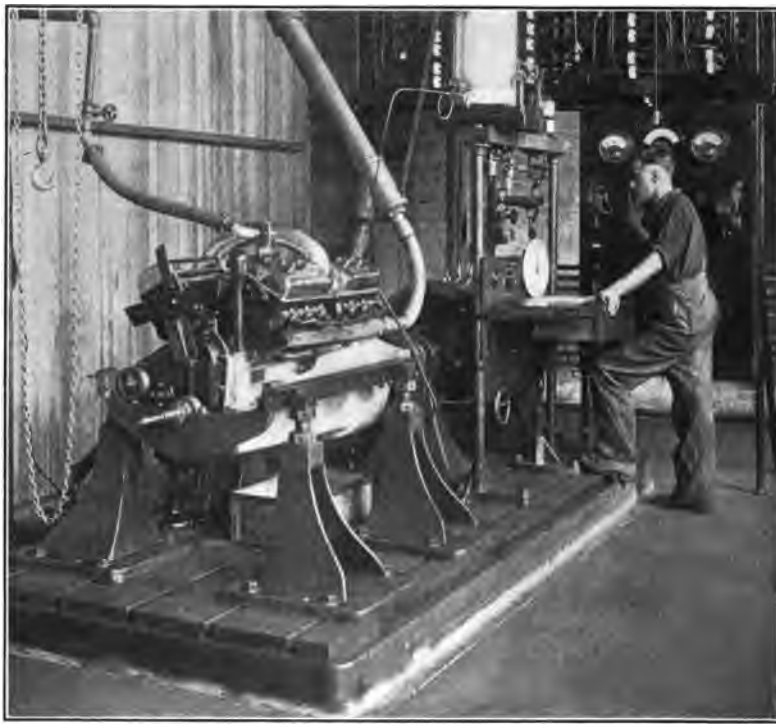


FIG. 125.—Engine coupled to Sprague electric dynamometer.

When ready for the test, the dial load-switch was set to the point corresponding to lowest external resistance, and hence heaviest current and heaviest counter-torque. The engine throttle was opened, and the field rheostat then turned to the left to strengthen the field. This gradually put so heavy a load on the engine that it was "pulled down" to the lowest speed on the curve, 400 r.p m. At this point the pull on the scales was measured and recorded, see attached table. With the throttle

still wide open, the field rheostat handle was turned a little to the right, lowering the counter-torque a little, so that the engine rose in speed to 500 r.p.m., where the pull on the scales was again measured. The rheostat handle was again turned, allowing the speed to rise to 600 r.p.m., and the pull again measured. In this way all the readings were taken and the curves plotted.

It is good practice to let the speed rise a little above measuring point, say, to 510, 615, etc., and then settle back to even speed, before attempting to measure the pull. The speed should be held steady long enough to insure simultaneous readings of torque and speed.

It is usually necessary in making curves on engines where the speed range is very wide to make one or two adjustments of the external resistance by the dial-switch. In the curve shown this would be made at about 800 r.p.m., that is, one setting would take the range from 400 to 800, and another from 800 to 3200. So practically the whole curve is made "on the field rheostat." This rheostat has a large number of steps, and there is an auxiliary plate which is "interpolated" giving steps between the steps of the main plate, thereby rendering possible very fine adjustments.

TABLE XII.—READINGS FROM WHICH THE CURVES (FIG. 124) OF THE PACKARD TWIN-SIX MOTOR (3" × 5") WERE PREPARED

R.P.M.	Scale reading in lb. (Arm length 21.008")	H.P. = $\frac{\text{lb.} \times \text{R.P.M.}}{3000}$
400	150	20
500	147	24½
600	145	29
700	142	33½
800	141	38½
1000	140¼	46¾
1200	138	55
1400	136	63½
1600	133	71
1800	132	79½
2000	125	83½
2200	120	88
2400	113½	91
2600	106½	92½
2800	100	93
3000	92¼	92¼
3200	84	89½

Figure 125 shows an engine on a stand attached to a Sprague "Electric Cradle Dynamometer," with scales and switchboard in plain view.

Figure 126 shows the test curves of the Class B Military Truck Engine, frequently referred to in previous chapters. These curves have been taken on the standard S. A. E. Testing

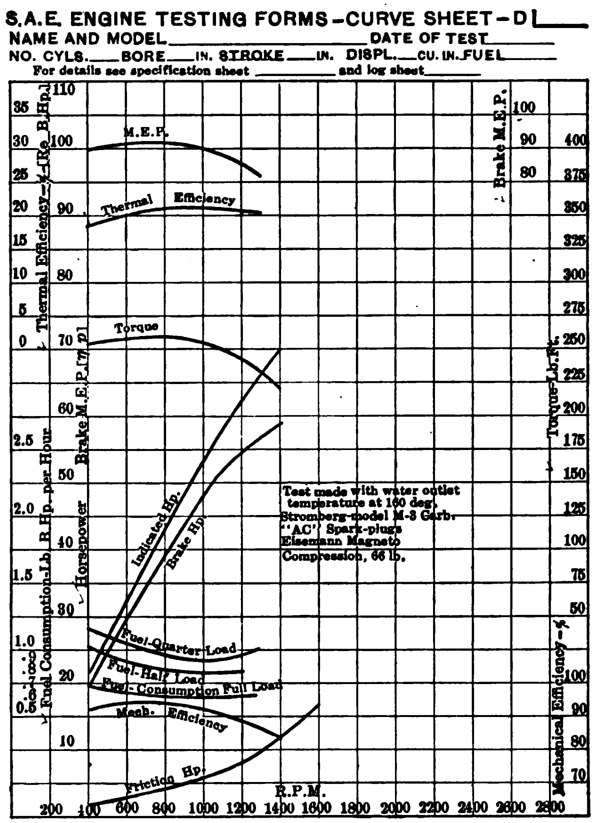


FIG. 126.—Test curves of U. S. A. Class B military truck engine.[1]

Forms and show the tests recommended by the S. A. E. In previous chapters it was described how the tests of the curves here shown, are found, with the exception of the friction horsepower test. The friction of a motor is easily measured with an electric dynamometer.

[1] Test Curves of the U. S. A. Class B Military Truck Engine, from the Journal of the Society of Automotive Engineers, Jan., 1918, by A. F. Milbrath.

If d is the length of the torque arm (see Chapter XX), R the r.p.m. and W the weight in lb. indicated on the scale, then the friction horsepower $= \frac{dRW}{5252.1}$. With a length of arm of 21.008 inches, the friction horsepower would simply be weight in pounds indicated on the scales multiplied by r.p.m. and divided by 3000 (see notes on friction horsepower tests in the following rules).

RULES AND DIRECTIONS RECOMMENDED BY THE S. A. E. ACCORDING TO THEIR ENGINE TESTING FORMS[1]

Specification Sheet—A

(See corresponding numbers on sheet—B)

(3) The compression volume is the volume occupied by the charge when the piston is at the top of the compression stroke. To measure this volume, with the piston on dead center at end of compression stroke (*i.e.*, with both valves closed) fill the compression space from a graduate containing a known volume of light oil or kerosene. Care must be taken to correct for leakage. Total volume of cylinder = piston displacement + compression volume. Give compression pressure at 100 to 120 r.p.m., or at speed of standard starter.

(4) State number of cylinders cast integral, whether offset, type of cylinder head, whether water space is provided between adjacent cylinders.

(6) State whether water or air-cooled. If the former, state whether pump or thermosyphon. Note if two pumps or thermostat are used. State type of pump.

(7) Weight of piston with rings and pin should include weight of bushings, screws, or other piston-pin fastening devices in the piston. Record all weights in pounds and decimal parts thereof. In measuring length of piston and distance from center of pin to top of piston, deduct any chamfer or crown at top of piston.

(8) Specify whether rings are concentric or eccentric; give name, sketch or description of special types. If oil-ring is used, state location.

(9) In giving weight of connecting-rod, include weight of all bushings, bolts, screws, and oiling devices normally attached to the rod. For piston-pin see (7). The connecting-rod must be horizontal while the ends are being weighed, the ends being supported by knife-edges or arbors. For V-type engines, state lower end construction.

[1] From the First Report of the Engine and Transmission Division adopted March, 1917, by the Society of Automobile Engineers.

(10) Under location, state whether in connecting-rod or piston.

(12) Diameters and lengths of bearings are to be stated in order from front to rear.

(14) Describe contour of cam, *i.e.*, uniform acceleration, tangential, etc.

(15 and 16) In case of non-poppet valves, describe and give dimensions.

(17) Reciprocating parts of directly operated poppet valves include valve, valve-lifter, valve-spring retainer and lock, and half of valve-spring.

(19) To determine valve-timing, mark top and bottom dead centers on flywheel rim; also points at which each valve opens and closes, engine cold and tappets set for standard clearance. Measure with flexible steel tape the length of arcs thus marked on flywheel. Reduce to degrees. Check both top and bottom dead centers for engines with offset cylinders.

(20) Moment of inertia of the flywheel is to be given in mass $\left(\frac{\text{weight in pounds}}{32.2}\right)$ and foot units. Moment of inertia is equal to the mass multiplied by the square of the radius of gyration. $I = MR^2$.

The complete weight of engine should include oil, water, and all mechanically attached units necessary for normal functioning of engine, such as carbureter and its attachments, magneto, ignition distributor, generator, starting motor, fan, governor, etc. Do not include such accessories as horn, tire-pump, vacuum tank, etc. List weight of each item separately.

(24) State if heated by water or exhaust, and whether part or all of the air entering carbureter is heated.

(25) Under "general principles of operation" give description; *e.g.:* "Venturi type with single adjustable nozzle and single auxiliary air-valve with one spring;" "straight-tube type, four non-adjustable nozzles coming into operation successively as air-flow increases."

(26) By description and sketch, give general form, approximate inside diameters of different portions, and specify which, if any, parts are jacketed.

(27) In case of two systems, state which was used in test.

(29) State if spark is fixed, or if spark control is automatic or manual. Maximum spark advance and retard are to be given in degrees of crankshaft rotation.

(30) Give material of insulation, number of sparking points on electrodes.

(31) In addition to exact location in combustion chamber, state whether vertical, horizontal or inclined, and whether plug extends into combustion chamber.

(32) Give the general type of lubrication system, *e.g.:* "recirculating

splash;" "force-feed and spray;" "complete force-feed." Then describe in detail action of system and course taken by oil. State oil pressures and type of pump used. State name and grade of oil used.

LOG SHEET AND CURVE SHEET

The Log Sheet and Curve Sheet have been designed for conveniently recording data and plotting curves covering the usual Standard Engine Tests. For special tests, these may be modified or special forms used.

GENERAL RULES AND DIRECTIONS

A complete Standard Engine Test includes the determination at different speeds of: (1) Max. hp.; (2) Fuel economy at max., at ¾, at ½ and at ¼ max. hp. at each of the speeds; (3) Friction-hp. From these determinations, the following curves are plotted on the Curve Sheet:

1. Torque—r.p.m.
2. Max. hp.—r.p.m.
3. Brake m.e.p.—r.p.m.
4. Friction-hp.—r.p.m.
5. Mechanical eff.—r.p.m.
6. Fuel per b.hp. per hour, max. hp. at each speed.
7. Fuel per b.hp. per hour, ¾ max. hp. at each speed.
8. Fuel per b.hp. per hour, ½ max. hp. at each speed.
9. Fuel per b.hp. per hour, ¼ max. hp. at each speed.
10. Thermal eff., max. hp. at each speed.
11. Thermal eff., ¾ max. hp. at each speed.
12. Thermal eff., max. hp. at each speed.
13. Thermal eff., ¼ max. hp. at each speed.

Emphasis is laid upon the value of the determination, in addition to the usual runs at max. hp. at each speed, of fuel consumption and thermal efficiency at each speed with the engine developing ¾, ½ and ¼ of its max. hp. *at that speed.* Automobile engines operate a large proportion of the time on part load, and in the study of the operating characteristics of engines, these curves are of great importance. During the max. hp. run at any speed, the max. torque (or load) is determined. For runs at ¾, ½ and ¼ max. hp. at the same speed it is only necessary to set the torque (or load) at ¾, ½ and ¼ the maximum for this speed and make proper throttle setting for such load.

During the complete test, control of engine shall be by means of throttle and spark only. Engine adjustments shall be made for best horsepower output (*i.e.*, carburetor setting, spark-plug gaps, etc.), and in no case are such adjustments to be changed during the complete test.

Test runs should not be made until the engine has been run-in sufficiently to show no appreciable difference in friction before and after a run of 30 min. at the speed of maximum torque with the throttle wide open.

Where test is to be made of a stock engine, all parts, accessories, lubri-

S. A. E. ENGINE TESTING FORMS—SPECIFICATION SHEET—B

Name and Model .... Date of Test ....
Manufacturer ....
*(1) General Type .... Cycle ....
(2) No. of Cyls. ...... Bore ...... in., Stroke ...... in., Piston Displ. per Cyl. ...... cu. in., Total ...... cu. in.
(3) Compression Vol. (V_c) ...... cu. in., Total Vol. of Cyl. (V) ...... cu. in., Compression Ratio = $\frac{\text{Comp. Vol.}}{\text{Total Vol.}} = \frac{V_c}{V} =$ ......
Compression Pressure ...... lb. gage at ...... r.p.m.
(4) Type of Cyl. Casting .... Matl. ....
(5) Type of Valves .... Location ....
(6) Cooling System ....
(7) Piston, Type .... Matl. ....
Wt. with Rings and Pin ...... lb., Length ...... in., Distance Center of Pin to Top of Piston ...... in.
(8) Piston-rings, No. per Piston ...... Type .... Width ...... in.
(9) Connecting-rod, Type ....
Length, c. to c. ...... in. Weight, Upper End ...... lb., Lower End ...... lb., Total ...... lb.
(10) Piston-rod Bearings, Diam. ...... in., Total Length ...... in., Matl. ...... Location ......
(11) Connecting-rod Bearings, Diam. ...... in., Length ...... in., Matl. ...... Type ......
(12) Crankshaft Bearings, No. ...... Diams. ......
Material ...... Lengths ......
(13) Camshaft Bearings, No. ...... Diams. ......
Material ...... Lengths ......
(14) Type of Cams ...... Type of Valve-lifters ......
(15) Inlet Valves, No. per Cyl. ...... o.d. ...... in., Port Diam. ...... in., Lift ...... in., Seat Angle ...... deg.
(16) Exhaust Valves, No. per Cyl. ...... o.d. ...... in., Port Diam. ...... in., Lift ...... in., Seat Angle ...... deg.
(17) Weight of Valve Reciprocating Parts, Inlet ...... lb., Exhaust ...... lb.
(18) Valve-spring Tension, Inlet Open ...... lb., Closed ...... lb., Exhaust Open ...... lb., Closed ...... lb.
(19) Valve-timing, Inlet Valve Opens ...... deg. ...... Top Center, Closes ...... deg. after Lower Center
Exhaust Valve Opens ...... deg. before Lower Center, Closes ...... deg. ...... Top Center
(20) Flywheel, o.d. ...... in., Weight ...... lb. Moment of Inertia ......
(21) Weight of Engine ...... lb., Including ......
....

Carburetion

(22) Carbureter, Name and Model .... Nom. Size ...... in.
(23) Specifications (Size of Nozzles, etc.) ....
(24) How Heated ....
(25) General Principles of Operation ....
....
(26) Description of Intake Pipe ....
....

Ignition

(27) Name and Type of System ....
(28) Type of Distributor .... Firing Order ....
(29) Type of Breaker ...... Maximum Spark Advance ...... deg., Retard ...... deg.
(30) Spark-plugs, Name and Type ...... Size ...... in.
(31) Location ...... Gap ...... in.

Lubrication System

(32) Type and Description ....
....
....

*See Notes Corresponding to Numbers under "Rules and Directions."

Form from the First Report of the Engine and Transmission Division adopted March, 1917, by the Society of Automobile Engineers, New York.

S. A. E. ENGINE TESTING FORMS

LOG SHEET—C

NAME AND MODEL..
No. CYLS........BORE.......in. STROKE.......in. DISPL. (D).....cu. in.

FUEL..........B.t.u. per Lb.........Sp. Grav.........at..........°F.
Dynamometer..................................Arm (R)......................ft.
Room Temp......°F. Bar......in. Hg. Humidity..........................
Laboratory.................................Date..............................
Observers..

	Run Number	Symbol	Formula	1	2	3	4	5	6	7	8	9	10	11	12	13	14
BRAKE HORSEPOWER AND FUEL CONSUMPTION	Time Started		*														
	Duration of Run—min.	t	*														
	Counter Start	C_0	*														
	Counter Finish	C_t	*														
	Total Rev.	r	$C_t - C_0$														
	Av. r.p.m.	N	$\frac{r}{t}$														
	Brake Load at Arm R	P	*														
	Torque in lb. ft.	T	PR														
	Brake m.e.p.	np	$\frac{150.8T}{D}$														
	Brake hp.	B.HP.	$\frac{PRN}{5252.1}$														
	Friction hp. at N	F.HP.	F.HP. Curve														
	Indicated hp.	I.HP.	B.HP.+F.HP.														
	Mechanical Efficiency	M.E.	$\frac{\text{B.HP.}}{\text{I.HP.}}$														
	Temp. Jacket Water—In		*														
	Temp. Jacket Water—Out		*														
	Temp. Air to Carb.		*														
	Wt. Fuel, Start	W_0	*														
	Wt. Fuel, Finish	W_t	*														
	Lb. Fuel Used	W	$W_0 - W_t$														
	Lb. Fuel per b.hp. hr.	F	$\frac{60W}{t \times \text{B.HP.}}$														
	Thermal Eff. re b.hp.	T.E.	$\frac{2545}{F \times \text{B.T.U.}}$														
FRICTION HP.	Time Started		*														
	Duration of Run—min.	t	*														
	Counter Start	C_0	*														
	Counter Finish	C_t	*														
	Av. r.p.m.	n	$\frac{C_t + C_0}{t}$														
	Brake Load at Arm R	p	*														
	Friction hp.	F.HP.	$\frac{pRn}{5252.1}$														
	Mean Temp., Jacket Water		*														

* Laboratory Readings. See also Specification Sheet..........and Curve Sheet..........

cants, etc., must be stock. In every case, all regular equipment must be on the engine and operating (*e.g.*, fan, generator, etc.).

Before beginning any run, the engine should be brought to a condition of sustained operation under the conditions of the run and it is imperative that in every case r.p.m., brake loads, rate of fuel consumption, cooling-water temperatures, oil temperature, air draft, etc., remain substantially constant, steady and sustained throughout the run. Flash readings and tests are unscientific and misleading.

Number of Runs.—In every test, enough runs should be taken throughout the speed range so that the points therefor when plotted will indicate clearly the shape and characteristics of the curves. For horsepower and fuel economy tests, it is recommended that runs be made at intervals of approximately 200 r.p.m. A run should be made at the lowest steady operating speed of the engine. All points from which curves are plotted are to be clearly shown on the Curve Sheet.

Duration of Runs.—The duration of Brake-horsepower tests shall not be less than 3 min. Where Fuel Consumption is measured, the duration of tests shall not be less than 5 min. The duration of Friction-horsepower tests shall not be less than 1 min. The above stated times are minima. In most instances it is desirable to make the runs of longer duration, and it is imperative that in every case r.p.m., brake loads, rate of fuel consumption, cooling-water temperatures, oil temperature, air draft, etc., remain substantially constant and steady throughout the run.

Balancing Dynamometer.—Before any readings of Brake Load are recorded, great care should be exercised to see that the dynamometer itself is properly balanced. For the electric-cradle type of dynamometer this balancing is accomplished as follows:

The dynamometer is run idle as a motor (drawing current from the line) and a suitable counter-balance on the field frame—which should be perfectly free to turn within limits in ball-bearing trunnions—is then adjusted so that the platform scales read zero. This reading should be obtained with the dynamometer rotating first in one direction and then in the other. The reaction of the armature on the field frame will exactly balance the friction of the brushes and armature bearings carried in the field frame. With the armature still rotating, check-weights (or pieces of metal having a known weight) should be hung from the knife-edge on the dynamometer arm. If the reading recorded by the platform scales is equal to the known weight applied, the dynamometer can be considered as balanced.

Brake Loads.—Readings for Brake Loads should be taken with accurately calibrated platform or beam scales. The connection of the dynamometer arm to these scales by means of knife-edges, calibrated spring balance and tripod or suitable linkage is recommended. Suitable counter-balances or tare loads must be accurately provided. The spring balance gives a quick approximate reading for Brake Load; it serves to cushion the platform or beam scales from shock and vibration. During any run, the platform or beam scales are kept balanced, and the loads registered thereby must be substantially constant and steady throughout the run.

Revolutions Per Minute.—Speed in revolutions per minute should be invariably taken from positively driven counters which engage at the

beginning of the run and disengage at the end. The difference between the two readings, divided by the duration of run in minutes, then gives the true average speed. Tachometers, even though carefully calibrated, are not sufficiently reliable for r.p.m. readings. In connection with the speed counters mentioned, however, the tachometer may be used as an approximate check on average speed, also as an indicator of variations in speed before or during the run.

It is recommended that the maximum allowable variation in speed during a run shall be 50 r.p.m.

Fuel Consumption.—The method recommended for measuring fuel consumption is by noting the decrease in weight of a tank from which fuel is being fed to the carburetor. The tank should be placed on sensitive platform scales at a proper level above the carburetor, and connected to the fuel-supply pipe by a short horizontal length of rubber tubing. This tubing should be very flexible and should not be drawn taut, to avoid interference with the weighing. Weighings should be made as follows:

Set counterpoise so that scale-beam will fall just as run is started. Note the setting and the time at which the scale-beam falls. Move the counterpoise back to the next pound mark, or to such a point that it will fall just before the end of the run, and note carefully the time when beam again falls. From the difference between the two times and the two weights recorded, the fuel consumption per hour can be readily determined.

The counterpoises may be successively set back for small quantities (say 1/4 or 1/2 lb.) and the times noted during the progress of the run. This gives an indication of the steadiness of fuel consumption throughout the run, and in no wise interferes with the major measurements outlined in the previous paragraph.

Temperatures.—All temperatures are to be given in degrees Fahrenheit.

A reliable glass straight-stem thermometer should be placed near the carburetor air-inlet in order to measure the temperature of the entering air. This thermometer should be read at least three times during each run, one of these times to be at beginning and one at end of run.

Thermometers should be placed also in suitable wells or sockets, one near the inlet of the pump and another as close as possible to the water-outlet of the engine. These wells or sockets should be in pipes that run full, so that water continually circulates about them. They should be filled with oil or mercury, and careful readings taken at least three times during each run, one of these times to be at beginning and one at end of run.

In order to afford a fixed basis of comparison, it is recommended that the outlet water temperature for engines be kept at 175°F. (+ or − 5°). Control of the outlet temperature can be accomplished by thermostat located in the outlet line or by external control of quantity or temperature of inlet water. Where thermostat or other cooling water regulating devices are standard upon an engine, these may be attached and operating during a test.

In every case, inlet and outlet cooling-water temperatures should remain substantially constant and steady throughout a run. It is recommended

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that the maximum allowable variation in cooling-water temperature shall be 10°F.

During friction-horsepower runs it is desired to obtain the mean temperature of the jacket water. If the water is pump-circulated, the average of the inlet and outlet temperatures may be taken. If thermosyphon circulation is used, the water will not circulate noticeably during a friction-horsepower run. The mean jacket-water temperature for such engines can be taken by inserting thermometers into the jacket space, the average of readings being taken. In every case of friction-horsepower test, the test must be made immediately after the corresponding brake-horsepower test, before the engine has cooled.

An air draft should be provided which approximates in amount and effect the air draft on the road with the car moving at a speed corresponding to the given engine speed. During friction-horsepower tests, of course, this air draft is shut off, in order not to cool the engine.

For air-cooled engines, the air draft is of the greatest importance.

Friction-horsepower.—The approximate friction-horsepower of an engine can be measured best by means of an electric dynamometer, preferably of the cradle type. The dynamometer is used to drive the engine under test at various speeds, and the torque reaction is measured. This will be in the opposite direction to that obtaining while the engine is driving the dynamometer, so that provision must be made for measuring the torque on both sides of the dynamometer, or else suitable linkage must be provided to change the direction of the pull. The test for friction-horsepower should be made immediately after the brake-horsepower test, before the engine has cooled, in order to keep the condition of the lubricating oil and the friction of the parts the same as during the brake-horsepower test, as nearly as possible. During this test the throttle of the engine should remain in the same position as for the corresponding brake-horsepower test. Compression-relief cocks should remain closed and all accessories, such as magneto, generator, pumps, etc., used during the brake-horsepower test, should be in operation. See notes on friction-horsepower runs under the various headings.

Indicated-horsepower.—Approximate indicated-horsepower is obtained by adding to the brake-horsepower at any given speed the friction-horsepower obtained at the same speed.

If the friction-horsepower and brake-horsepower tests are not made at exactly the same speeds, the friction-horsepower at any given speed can be obtained from the friction-horsepower curve plotted on the Curve Sheet. Tedious interpolation is thus avoided.

Fig. 126 shows curve sheet (D) recommended by the Society of Automotive Engineers.

CHAPTER XXIV

BRAKE-HORSEPOWER TEST BY HYDRAULIC DYNAMOMETER

Another convenient method for testing the Brake Horsepower of a motor is with a Hydraulic Dynamometer. See Fig. 127 which is the "Spillman Hydraulic Dynamometer."

This Dynamometer has a double vane turbine mounted on a shaft to revolve between two fixed turbine heads. Water enters these heads, preferably under pressure. One end of the shaft may be attached by a universal coupling to the engine or other device under test, and as the shaft revolves the water is passed back and forth between the revolving and fixed turbines and causes the dynamometer, mounted on roller bearings, to tilt in the direction of the rotation.

Two arms of suitable length are attached to the dynamometer body. From one of these arms is suspended a direct reading spring scale, which indicates the pull in pounds. The opposite arm is provided with a sliding counter-weight by which the dynamometer is balanced before connecting it to the engine.

The handwheel and geared gates control the volume of water which thrashes back and forth between the rotating and fixed turbines. There is no internal friction in the brake, all the load being taken care of by the reaction between the turbines. The volume of water within the brake regulates the load imposed upon the mechanism. For maximum load the gates should be wide open and inversely for no load. As all of the power absorbed in the brake is converted into heat, it is necessary to introduce cooling water by means of the inlet globe valve provided.

It is found that where city water pressure is connected directly to the dynamometer it is possible to control the load very largely by opening and closing the inlet globe valve after the gates have been set by handwheel in that position which practice will show to be most desirable.

In the absence of water pressure, the dynamometer may be supplied from a barrel connected at the bottom with the brake

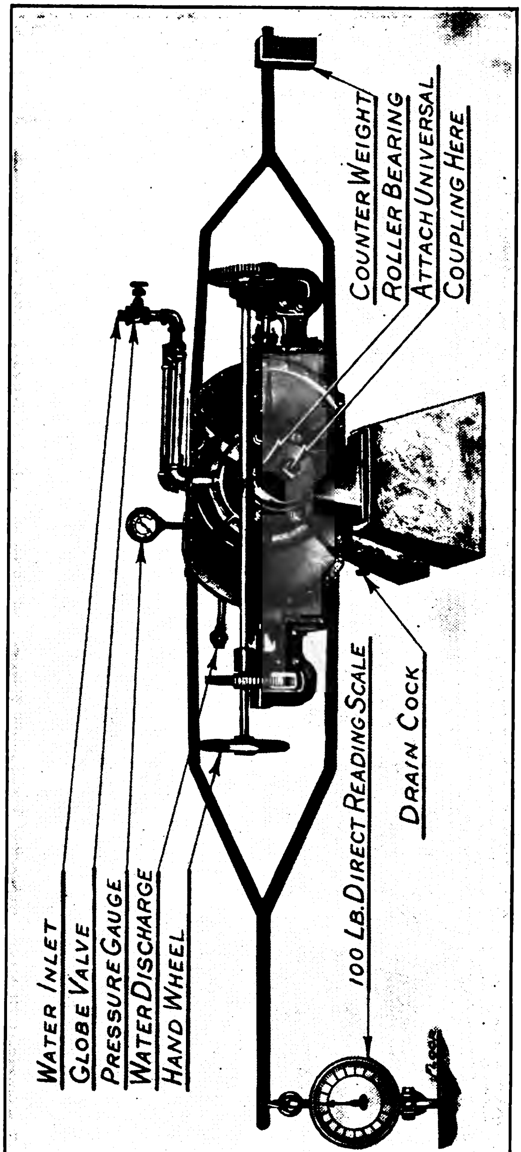

FIG. 127.—Herschell-Spillman hydraulic dynamometer.

intake and the discharge returned to the barrel. The brake will develop suction sufficient to feed itself, but cooling water must be introduced as required.

The dynamometer may be mounted on any supports suitable to the testing conditions.

OPERATING THE DYNAMOMETER

Mount the apparatus so it moves freely upon the supporting roller bearings. Attach scale to clevis arm and to the floor so that the dynamometer stands level. Balance the dynamometer by moving sliding counter-weight to proper position to make scale pointer read zero. Couple engine with good universal joint to square end of dynamometer shaft; to opposite end of shaft attach tachometer or other speed-indicating device. Start motor and let it run light, that is, without any water in dynamometer. Close drain cock, open discharge valve, open water inlet valve, open gates full by turning handwheel anti-clockwise. Engine will slow down as dynamometer takes up load. Close discharge valve to allow passage of sufficient water to maintain even temperature. Open engine throttle and engine should speed up. If speed does not increase dynamometer is too heavily loaded, open discharge valve and close intake valve correspondingly. To load engine close discharge valve and open intake valve. If a greater load is desired than is obtained by manipulation of water inlet and outlet as above, turn handwheel clockwise to reduce load and anti-clockwise to increase load. A little practice will determine whether or not to change the handwheel or the inlet and discharge valve settings. The brake does not work instantaneously, it will take probably thirty seconds to stabilize the dynamometer.

The pressure gauge will indicate whether the brake is full of water and producing sufficient pressure.

The drain cock provides for drainage of water to prevent freezing.

Reading the Load.—This dynamometer has a constant of .001 and the hydraulic and mechanical friction are negligible so that to determine the amount of power being absorbed by the brake, it is necessary only to multiply the revolutions per minute by the number of pounds indicated on scale and point off three places in the product: Thus with speed of 1000 r.p.m. and 20 lb. indicated on scale, the dynamometer is absorbing 20 hp.

CHAPTER XXV

TRACTIVE EFFORT AND HORSEPOWER REQUIRED FOR MOTOR VEHICLES

Before determining the actual H.P. required for driving a vehicle it is necessary to know the maximum load to be carried on the four wheels, *i.e.*, the weight of car plus useful load. When a motor vehicle is propelled on a level road, it has to overcome first of all, the rolling resistance of the ground (the tractive resistance), where the wheels contact the road; secondly, the friction between the engine crankshaft and the road wheels, *i.e.*, the friction in the transmission gear and the rear axle; and thirdly, the air resistance. (The air resistance is very small unless the motor vehicle travels at high speed, for this reason it will not be included in our calculations at first, but it will be mentioned later.)

When the car is propelled up an incline, additional power is required, to lift the weight of the vehicle so to say, or to pull it up the hill, in addition to the tractive resistance on level road.

The road resistance depends upon the quality and the character of the road surface as well as on the diameter of the wheels. A large wheel for instance will not enter all the small holes on the road like a smaller wheel, and consequently, in going over ruts, the axle with a larger wheel will not be raised or lowered to the same extent as an axle with a smaller wheel. In practice, it is difficult to separate the various resistances; since the friction in the rear axle and transmission gear is comparatively small, all the losses will be included in the road resistance, when figuring the necessary H.P.

If a motor vehicle is pulled very slowly and the amount of pull required (as for instance, pulling the vehicle by a spring balance) is measured in pounds, the actual pull so found is called the tractive effort, and the pull required per unit weight, say, for example, the pull per ton, is frequently called the traction coefficient.

For smooth macadam road or hard, smooth asphalt pavements

the effort required to pull the car is about 22 lb. per ton of weight (in cars equipped with rubber tires).

The tractive effort per ton to overcome road resistance for various road surfaces is approximately as follows:

Hard smooth asphalt	22 lb.
Wood paving	26 lb.
Good macadam	35 lb.
Bad macadam	45 to 90 lb.
Cobbles	55 lb.
Bad cobbles	up to 220 lb.
Sand road	from 325 to 500 lb.

For practical calculations assume the average road resistance as 35 lb. and the friction losses in the car as 15 lb. per ton or a total of 50 lb. per ton, over good macadam roads. If the total weight of the car in pounds, including the load $= W$; the total effort required to overcome the resistance, in pounds $= \dfrac{W \times 50}{2000}$ the tractive effort.

If the car is driven up a grade g of say 7 in one 100, which is $\dfrac{7}{100} = .07$, usually called a seven (7%) per cent. incline, an additional effort is required.

Our formula then becomes tractive effort due to grade $= W \times$ grade, and tractive effort due to grade per ton $= 2000 \times$ grade $= \dfrac{W\ (\text{Road resistance} + 2000 \times \text{grade})}{2000}$; in the above examples, therefore,

the total tractive effort $= \dfrac{W\ (50 + 2000 \times .07)}{2000}$.

If we call m the speed of the car in miles per hour, then the speed in feet per minute is $= \dfrac{m \times 5280}{60} = 88m.$

The H.P. required to overcome the traction resistance on a level road is

$$\text{H.P.} = \frac{W \times R \times 88m}{2000 \times 33{,}000}$$

while the total power necessary, including that for grade is

$$\text{H.P.} = \frac{W\ (R + 2000 \times g)88m}{2000 \times 33{,}000} = \frac{W\ (R + 2000 \times g)m}{750{,}000}$$

$\left(\text{It should be remembered that } g \text{ is expressed in } \dfrac{\text{per cent.}}{100}\right)$

As an example, find the actual H.P. required to propel the car at 20 miles per hour on a level road, and up a 7 per cent. grade when the total weight of car is 5000 lb. and the road resistance per ton (including friction) 50 lb.

On a level road we have

$$\text{H.P.} = \frac{5000 \times 50 \times 20}{750{,}000} = 6.6$$

On a 7 per cent. grade we have

$$\text{H.P.} = \frac{5000 \times (50 + 2000 \times .07)\ 20}{750{,}000} = \frac{5000 \times (50 + 140)\ 20}{750{,}000} = 25.2$$

Suppose the vehicle should travel at 30 miles per hour up a grade of 1 in 7, or $\frac{1}{7} = .142$ or 14.2 in one 100.

$$\text{H.P.} = \frac{5000 \times (50 + 2000 \times .142)\ 30}{750{,}000} = 66.8$$

It is interesting to know the power necessary to overcome wind resistance; this is comparatively small except at high speeds. First, it is necessary to know the wind pressure at various velocities per unit area. From experiments conducted by the U. S. Signal Service at Mount Washington, N. H., the following was ascertained:

TABLE XII.—WIND PRESSURE

Velocity in miles per hour	Pressure, pounds per square foot
10	0.4
20	1.6
30	3.6
40	6.4
50	10.0
60	14.4
80	25.6
100	40.0

In studying these values, it is found that they follow approximately the law, $P = .004m^2$, where m is the velocity in miles per hour.

If A is the projected front area of the body of the vehicle in

square feet, and V the speed in feet per minute, then the H. P. required to overcome air resistance is

$$\text{H.P.} = \frac{PAV}{33{,}000} = \frac{.004m^2VA}{33{,}000} = \frac{.004m^2A\ 88m}{33{,}000} = \frac{m^3A}{93{,}750},$$ for

practical calculations it is sufficiently accurate to use H. P. $= \frac{m^3\,A}{100{,}000}$.

Suppose the vehicle is traveling at 20 miles per hour, and the area of the windshield and the radiator (and other surfaces exposed to direct air current) is 10 sq. ft., then

$$\text{H.P.} = \frac{20 \times 20 \times 20 \times 10}{100{,}000} = \frac{80{,}000}{100{,}000} = .8 \text{ H.P.}$$

At 60 miles per hour the H.P. would be:

$$\text{H.P.} = \frac{60 \times 60 \times 60 \times 10}{100{,}000} = \frac{2{,}160{,}000}{100{,}000} = 21.6 \text{ H.P.}$$

The complete formula for the required H.P. including all the losses thus becomes

$$\text{H.P.} = \frac{W(R + 2000 \times \text{grade})m}{750{,}000} + \frac{m^3A}{100{,}000}$$

The tractive effort, in addition to the character of the road surface, is largely influenced by the kind of tires which are used on the motor vehicle; for pneumatics properly inflated, or for good solid rubber tires, running at low speeds, and not overloaded, the figure given before, *i.e.*, 50 lb. per ton, will be found sufficiently accurate for practical purposes.

CHAPTER XXVI

TRACTIVE FACTOR, TORQUE, ETC.[1]

In the last Chapter the tractive effort and the H.P. required for motor vehicles was considered.

In this Chapter some features of the specifications of the Class B motor trucks, which specifications were prepared by the Quartermaster's General's Office, U. S. Army, will be examined.

The specification contains the following clauses:

Tractive Factor on high gear to be not less than .0775 (which was subsequently changed to .063).

Tractive Factor on low gear, .338 (which was subsequently changed to .307).

Worm gear reduction in the rear axle, 9.5.

Low gear reduction in transmission case 5.93 (in addition to worm gear reduction).

Worm gear to have transmission efficiency, 85 per cent.

Total transmission efficiency, including low gear, 70 per cent. It was estimated that the weight of the chassis would be 8000 lb., and the weight complete, including the load 16,500 lb.

It was also determined that the engine required, was to develop a torque of not less than 2700 pound-inches.

The Tractive Factor means simply the tractive effort in lb. divided by the total weight in lb. on the tires, that is

$$\text{Tractive Factor (T.F.)} = \frac{\text{Tractive Effort (T.E.)}}{\text{Total Weight (W) in lb.}}$$

The torque (t) of an engine is the effort, or pull in lb. exerted by the crankshaft at a certain radius. Knowing the H.P. of an engine, it is easy to determine the torque on the shaft. Take the H.P. of a motor, at the piston speed of 1000 ft. per minute, which is the power of an engine according to the N.A.C.C. rating.

The torque of the motor in pounds at 1 inch radius =

$$\frac{33{,}000 \times \text{H.P.} \times 12}{2 \times \pi \times R} = t \text{ pound inches.}$$

R represents the revolutions per minute. To have the H.P. translated into inch pounds we had to multiply it by 33,000 × 12.

[1] See C. T. Myers in the Journal of Society of Automotive Engineers, January 19, 1918, also S. A. E. Transactions, 1914, Part II, from which this chapter was largely taken.

If L is the stroke in inches,

$R = \frac{1000 \times 12}{2L}$ (remembering that the power in our case is developed at a speed of 1000 ft. per minute, and the H.P. is $\frac{D^2N}{2.5}$, where D is the diameter of the cylinder and N the number of cylinders).

By substituting the values for R and for H.P. in the formula for torque, we may write

$$t = \frac{\frac{33{,}000 \times D^2 \times N \times 12}{2.5}}{\frac{2\pi \times 1000 \times 12}{2L}} = \frac{33{,}000D^2N \times 12 \times 2L}{2.5 \times 2\pi \times 1000 \times 12}$$

By simplifying we have $t = \frac{33D^2NL}{2.5\pi} = 4.2D^2NL.$

The Tractive Effort (T.E.) at the periphery of the driving wheels,

$$\text{TE} = \frac{t \times e}{\frac{1}{2}d} = \frac{2te}{d}$$

where e is the gear reduction and d the diameter of the driving wheels. In order to be accurate this formula must also be multiplied by the efficiency of the transmission as we shall see later.

Substituting the values for t found before, we have

$$\text{TE (in lb.)} = \frac{2 \times 4.2D^2NLe}{d} = \frac{8.4D^2NLe}{d}$$

The Tractive Factor, then $= \frac{\text{TE}}{W} = \frac{8.4D^2NLe}{dW}$

This equation being based on the assumption that the motor develops its torque at a speed equivalent to its N.A.C.C. rating, and with a transmission efficiency of 100 per cent.

In practice the torque may be taken as 1.2 times the values corresponding to the N.A.C.C. rating, and the efficiency of the transmission system is not 100 per cent., but varies between 70 per cent. and 85 per cent. If Et is the torque efficiency of the engine as compared with the N.A.C.C. rating, and Em is the mechanical efficiency of the transmission system from the engine crankshaft to the rear, we have

$$\text{TF} = \frac{8.4D^2NLe}{dW} \times \text{Et} \times \text{Em},$$

which is the resistance per lb. of weight to be overcome.

In the specification for Class B trucks, the rear wheels are equipped with tires of 40 inches diameter, and as mentioned before, the minimum engine torque was determined upon as 2700 pound inches.

While the efficiency on the high gear (with gear reduction in rear axle 9.5) was to be 85 per cent. = .85, and the low 70 per cent. = .70 (with gear reduction in transmission 5.93).

Therefore, TF for high gear $= \dfrac{2 \times 2700 \times 9.5 \times .85}{40 \times 16{,}500} = .066$

and for low gear TF $= \dfrac{2 \times 2700 \times 9.5 \times 5.93 \times .70}{40 \times 16{,}500} = .323$

If it is desired to determine the engine torque from the engine dimensions, we must multiply the result by 1.2, as explained before. Where the torque is specified, this, of course, is unnecessary, and it (Et) was therefore omitted from the foregoing equation.

In the last Chapter it was shown that the traction coefficient or the drawbar pull, to overcome road resistance, is about 35 lb. per ton, or $= \dfrac{35}{2000} = .0175$ lb. per lb. of weight.

With heavy trucks it is well to figure that it should be able to climb a grade of at least 3 per cent. on high gear, which is $\dfrac{3}{100} = .03$, thus the total TE per lb. is $.03 + .0175 = .0475$.

In the Class B truck engine, the bore is $4\frac{3}{4}$ inches and the stroke 6 inches. In order to determine the TF from the engine dimensions, we have the formula given before.

$$\text{TF} = \frac{8.4D^2NLe \times \text{Et} \times \text{Em}}{dW}$$

$$\text{for high gear TF} = \frac{8.4 \times 4.75^2 \times 4 \times 6 \times 9.5 \times 1.2 \times .85}{40 \times 16{,}500} = .0667;$$

$$\text{for low gear TF} = \frac{8.4 \times 4.75^2 \times 4 \times 6 \times 9.5 \times 5.93 \times 1.2 \times .7}{40 \times 16{,}500}$$

$= .326.$

If it is desired to find the hill-climbing ability of the truck, simply deduct the rolling resistance, which, we have seen, is .0175. Thus $.0667 - .0175 = .0492$ which represents a hill-climbing ability of $.0492 \times 100 = 4.9$ per cent.

On the low gear the hill-climbing ability would be .326 − .0175 = .3085, or a 30.85 per cent. grade.

It was found that the actual weight of the trucks with full load was 17,300 lb., from which we find the TF on high gear, .063, and for low gear .307.

CHAPTER XXVII

MATERIALS

Before giving the various material specifications, as recommended by the Society of Automotive Engineers, some qualities of steel will be discussed as on this depends very largely the satisfactory working of a motor.

MODULUS OF ELASTICITY

The elasticity of steel, or of a material, is the resistance which it offers to change of form on the application of a load, combined with the power of returning to its original shape after the load is removed. The elastic limit of a material is reached when the load causes a permanent set, that is to say, when the piece of material does not recover its original form after the load is removed. When a piece of metal is not loaded beyond its elastic limit, the stress produced is directly proportionate to the strain, so that the stress divided by the strain is a constant quantity. This constant quantity is called the Modulus of Elasticity; therefore the modulus of elasticity $= \frac{\text{stress}}{\text{strain}}$. This modulus is practically the same for all kinds of steel and is approximately 30,000,000.

It may be stated that "load" is the combination of external forces acting on a piece of structure, and this is called the load on that piece. The effect of a load, however small, on a structure, is a change of form, this change of form is called "strain." The internal forces which are called into play in the material of a structure to resist the tendency of the load to produce strain, are called "stress."

If a bar having a cross section of A square inches, carries a load of W pounds, either in tension or compression, and if the modulus

of elasticity of the material of the bar is E pounds per square inch, then the strain S produced is given by the formula $S = \frac{W}{EA}$.

Changes in temperature will also create a strain in the material; as is well known, heat will cause a metal to expand and *vice versa*. If the strain S is caused by change of temperature, the force W, which will bring the bar back to its original length, is given by the formula $W = EAS$ and this is the force which must be applied to prevent the strain or change of length, as the temperature changes.

Let us work out an example. It is desired to find the increase in length of a steel bar 6 ft. long and 3/4 inch in diameter when it is pulled with a force of 8960 lb. Using the letters we used before, $W = 8960$ lb., the modulus of elasticity of steel $E = 30{,}000{,}000$. The area $A = .7854 \times 3/4 \times 3/4$. The strain $S =$ increase in length over original length for each 1 inch length of bar $= \frac{W}{EA} = \frac{8960}{30{,}000{,}000 \times .7854 \times 3/4 \times 3/4}$. Therefore the increase in length, or the total length of the whole bar $= \frac{8960 \times 6 \times 12}{30{,}000{,}000 \times .7854 \times 3/4 \times 3/4} = .0486$ inches, or a little over 3/64″.

The following are steels and alloys recommended by the Society of Automotive Engineers as adopted by the Society in 1915.

SPECIFICATION NUMBERS

A numerical index system has been adopted in the numbering of the metal specifications. This system renders it possible to employ specification numerals on shop drawings and blue prints, which numerals are in themselves partially descriptive of the quality of material covered by such number.

The first figure indicates the class to which the steel belongs; thus 1 indicates the carbon steel, 2 nickel, 3 nickel chromium, etc. In the case of the alloy steels, the second figure generally indicates the approximate percentage of the predominant alloying element. The last two or three figures indicate the average carbon contained in "points" or hundreds of 1 per cent.; thus 2340 indicates a nickel steel with approximately 3 per cent. nickel (3.25 to 3.75 per cent.) and .40 per cent. carbon (.35 per cent. to .45 per cent.) and 51120 indicates a chromium steel with about 1 per cent. chromium (.90 per cent to 1.10 per cent.) and .120 per cent. carbon (1.10 per cent. to 1.30 per cent.).

The basic numerals for the various qualities of steels herein specified are as follows:

Carbon steel	1
Nickel steels	2
Nickel chromium steels	3
Chromium steels	5
Chromium vanadium steels	6
Silico-manganese steels	9

SPECIFICATIONS FOR STEEL

The specifications indicate the desired chemical composition.

PHYSICAL PROPERTIES

In interpreting the physical property and tabulations given in the following data, these considerations should be borne in mind:

1. The figures given have been made as valuable as possible to the engineer by indicating what can be expected as the average product of a given composition when treated in the specified manner, in average sections prevailing in motor car work.
2. At the same time the data have been so chosen as to proctet makers of treated stock and parts from unreasonable demands. This has been done by taking figures low enough to be obtained with reasonable certainty when open market stock of medium to high grade is treated in commercially efficient equipment, controlled by commercially accurate instruments.

For the sake of simplicity only average minimum figures for tensile strength, elastic limit, reduction of area and elongation have been adopted; these figures are based upon the following assumptions, heat treatment being kept constant:

1. The lowest tensile strength and elastic limit are produced with steels at the bottom of a given range in carbon.
2. The lowest reduction in area and elongation are produced with steels at the top of a given range in carbon.

Thus, for 1035 steel, the tensile strengths and elastic limits given are the average minimum as of a steel containing 0.30 per cent. carbon; the reductions of area and elongations are the average minimum as of a steel containing .40 per cent. carbon.

True elastic limits are given because they are consistently lower than the corresponding yield points.

The figures for hardness are conventional averages for the whole range of compositions within any given specification. In general, the Brinell hardness figure is subject to fluctuations of plus or minus ten to fifteen points, the Shore (scleroscope) hardness of plus or minus five points.

CARBON STEELS

SPECIFICATION No. 1010

0.10 Carbon

This is usually known in the trade as soft, basic open-hearth steel. It is a material commonly used for seamless tubing, pressed steel frames, pressed

TABLE XIII.—CARBON STEELS

S. A. E. Specification No.	Carbon		Manganese		Phosphorus (maximum)	Sulphur (maximum)	Heat treatment
	Minimum and maximum	Desired	Minimum and maximum	Desired			
1010	0.05 to 0.15	0.10	0.30 to 0.60	0.45	0.045	0.05	Quench at 1500°F.
1020	0.15 to 0.25	0.20	0.30 to 0.60	0.45	0.045	0.05	A or B
1025	0.20 to 0.30	0.25	0.50 to 0.80	0.65	0.045	0.05	H
1035	0.30 to 0.40	0.35	0.50 to 0.80	0.65	0.045	0.05	H, D or E
1045	0.40 to 0.50	0.45	0.50 to 0.80	0.65	0.045	0.05	H, D or E
1095	0.90 to 1.05	0.95	0.25 to 0.50	0.35	0.04	0.05	F

TABLE XIV.—SCREW STOCK

S. A. E. Specification No.	Carbon	Manganese	Phosphorus (maximum)	Sulphur
1114	0.08 to 0.20	0.30 to 0.80	0.12	0.06 to 0.12

TABLE XV.—STEEL CASTINGS*

S. A. E. Specification No.	Carbon	Phosphorus (maximum)	Sulphur (maximum)
1235	As required for physical properties	0.05	0.05

*See notes on page 269.

TABLE XVI.—NICKEL STEELS

S. A. E. Specification No.	Carbon		Manganese		Phosphorus (maximum)	Sulphur (maximum)	Nickel		Heat treatment
	Minimum and maximum	Desired	Minimum and maximum	Desired			Minimum and maximum	Desired	
2315	0.10 to 0.20	0.15	0.50 to 0.80	0.65	0.04	0.045	3.25 to 3.75	3.50	G, H or K
2320	0.15 to 0.25	0.20	0.50 to 0.80	0.65	0.04	0.045	3.25 to 3.75	3.50	G, H or K
2330	0.25 to 0.35	0.30	0.50 to 0.80	0.65	0.04	0.045	3.25 to 3.75	3.50	H or K
2335	0.30 to 0.40	0.35	0.50 to 0.80	0.65	0.04	0.045	3.25 to 3.75	3.50	H or K
2340	0.35 to 0.45	0.40	0.50 to 0.80	0.65	0.04	0.045	3.25 to 3.75	3.50	H or K
2345	0.40 to 0.50	0.45	0.50 to 0.80	0.65	0.04	0.045	3.25 to 3.75	3.50	H or K

TABLE XVII.—NICKEL CHROMIUM STEELS

S. A. E. Specification No.	Carbon		Manganese		Phosphorus (maximum)	Sulphur (maximum)	Nickel		Chromium		Heat treatment
	Minimum and maximum	Desired	Minimum and maximum	Desired			Minimum and maximum	Desired	Minimum and maximum	Desired	
3120	0.15 to 0.25	0.20	0.50 to 0.80	0.65	0.04	0.045	1.00 to 1.50	1.25	0.45 to 0.75*	0.60	G, H or D
3125	0.20 to 0.30	0.25	0.50 to 0.80	0.65	0.04	0.045	1.00 to 1.50	1.25	0.45 to 0.75*	0.60	H, D or E
3130	0.25 to 0.35	0.30	0.50 to 0.80	0.65	0.04	0.045	1.00 to 1.50	1.25	0.45 to 0.75*	0.60	H, D or E
3135	0.30 to 0.40	0.35	0.50 to 0.80	0.65	0.04	0.045	1.00 to 1.50	1.25	0.45 to 0.75*	0.60	H, D or E
3140	0.35 to 0.45	0.40	0.50 to 0.80	0.65	0.04	0.045	1.00 to 1.50	1.25	0.45 to 0.75*	0.60	H, D or E
3220	0.15 to 0.25	0.20	0.30 to 0.60	0.45	0.04	0.04	1.50 to 2.00	1.75	0.90 to 1.25	1.10	G, H or D
3230	0.25 to 0.35	0.30	0.30 to 0.60	0.45	0.04	0.04	1.50 to 2.00	1.75	0.90 to 1.25	1.10	H or D
3240	0.35 to 0.45	0.40	0.30 to 0.60	0.45	0.04	0.04	1.50 to 2.00	1.75	0.90 to 1.25	1.10	H or D
3250	0.45 to 0.55	0.50	0.30 to 0.60	0.45	0.04	0.04	1.50 to 2.00	1.75	0.90 to 1.25	1.10	M or Q
X3315	0.10 to 0.20	0.15	0.45 to 0.75	0.60	0.04	0.04	2.75 to 3.25	3.00	0.60 to 0.95	0.80	G
X3335	0.30 to 0.40	0.35	0.45 to 0.75	0.60	0.04	0.04	2.75 to 3.25	3.00	0.60 to 0.95	0.80	P or R
X3350	0.45 to 0.55	0.50	0.45 to 0.75	0.60	0.04	0.04	2.75 to 3.25	3.00	0.60 to 0.95	0.80	P or R
3320	0.15 to 0.25	0.20	0.30 to 0.60	0.45	0.04	0.04	3.25 to 3.75	3.50	1.25 to 1.75	1.50	L
3330	0.25 to 0.35	0.30	0.30 to 0.60	0.45	0.04	0.04	3.25 to 3.75	3.50	1.25 to 1.75	1.50	P or R
3340	0.35 to 0.45	0.40	0.30 to 0.60	0.45	0.04	0.04	3.25 to 3.75	3.50	1.25 to 1.75	1.50	P or R

*Another grade of this type of steel is available with chromium content of .15 per cent. to .45 per cent. It has somewhat lower physical properties.

TABLE XVIII.—CHROMIUM STEELS

S. A. E. Specification No.	Carbon		Manganese		Phosphorus (maximum)	Sulphur (maximum)	Chromium		Heat treatment
	Minimum and maximum	Desired	Minimum and maximum	Desired			Minimum and maximum	Desired	
5120	0.15 to 0.25	0.20	*	*	0.04	0.045	0.65 to 0.85	0.75	B
5140	0.35 to 0.45	0.40	*	*	0.04	0.045	0.65 to 0.85	0.75	H or D
5165	0.60 to 0.70	0.65	*	*	0.04	0.045	0.65 to 0.85	0.75	H or D
5195	0.90 to 1.05	0.95	0.20 to 0.45	0.35	0.03	0.03	0.90 to 1.10	1.00	M, P or R
51120	1.10 to 1.30	1.20	0.20 to 0.45	0.35	0.03	0.03	0.90 to 1.10	1.00	M, P or R
5295	0.90 to 1.05	0.95	0.20 to 0.45	0.35	0.03	0.03	1.10 to 1.30	1.20	M, P or R
52120	1.10 to 1.30	1.20	0.20 to 0.45	0.35	0.03	0.03	1.10 to 1.30	1.20	M, P or R

*Two types of steel are available in this class, one with manganese .25 per cent. to .50 per cent. (.35 per cent. desired), and silicon not over .20 per cent.; the other with manganese .60 per cent. to .80 per cent. (.70 per cent. desired), and silicon .15 per cent. to .50 per cent.

TABLE XIX.—CHROMIUM VANADIUM STEELS

S. A. E. Specification No.	Carbon		Manganese		Phosphorus (maximum)	Sulphur (maximum)	Chromium		Vanadium		Heat treatment
	Minimum and maximum	Desired	Minimum and maximum	Desired			Minimum and maximum	Desired	Minimum	Desired	
6120	0.15 to 0.25	0.20	0.50 to 0.80	0.65	0.04	0.04	0.80 to 1.10	0.95	0.15	0.18	S
6125	0.20 to 0.30	0.25	0.50 to 0.80	0.65	0.04	0.04	0.80 to 1.10	0.95	0.15	0.18	S or T
6130	0.25 to 0.35	0.30	0.50 to 0.80	0.65	0.04	0.04	0.80 to 1.10	0.95	0.15	0.18	T or U
6135	0.30 to 0.40	0.35	0.50 to 0.80	0.65	0.04	0.04	0.80 to 1.10	0.95	0.15	0.18	R or U
6140	0.35 to 0.45	0.40	0.50 to 0.80	0.65	0.04	0.04	0.80 to 1.10	0.95	0.15	0.18	T or U
6145	0.40 to 0.50	0.45	0.50 to 0.80	0.65	0.04	0.04	0.80 to 1.10	0.95	0.15	0.18	U
6150	0.45 to 0.55	0.50	0.50 to 0.80	0.65	0.04	0.04	0.80 to 1.10	0.95	0.15	0.18	U
6195	0.90 to 1.05	0.95	0.20 to 0.45	0.35	0.03.	0.03	0.80 to 1.10	0.95	0.15	0.18	U

TABLE XX.—SILICO-MANGANESE STEELS

S. A. E. Specification No.	Carbon		Manganese		Phosphorus (maximum)	Sulphur (maximum)	Silicon		Heat treatment
	Minimum and maximum	Desired	Minimum and maximum	Desired			Minimum and maximum	Desired	
9250	0.45 to 0.55	0.50	0.60 to 0.80	0.70	0.045*	0.045	1.80 to 2.10	1.95	V
9260	0.55 to 0.65	0.60	0.50 to 0.70	0.60	0.045*	0.045	1.50 to 1.80	1.65	V

*Steel made by the acid process may contain .05 phosphorus.

steel brake-drums, sheet steel brake-bands and pressed steel parts of many varieties. It is soft and ductile and will stand much deformation without cracking.

In a natural or annealed condition, this steel has little tenacity and must not be used where much strength is required. This quality of material is considerably stronger after cold drawing or rolling; that is, its yield point, is raised by such working. This is important in view of the fact that many of the wire and sheet metal parts mentioned above are used in the cold-rolled or cold-drawn form.

It must not be forgotten that when this steel (so cold worked) is heated, as for bending, brazing, welding, or the like, the yield point returns to that characteristic of the annealed material. *This remark also applies to all materials that have an increased yield point produced by cold working.*

This material in a natural or annealed state does not machine freely. It will tear badly in turning, threading and broaching operations. Heat treatment produces but little benefit, and that not in strength but in toughness. It is possible to quench this grade of steel and put it in a condition to machine better than in the annealed state.

The heat treatment which will produce a little stiffness is to quench at 1500°F. in oil or water. No drawing is required.

Physical Characteristics

	Annealed	Cold rolled or cold drawn
Yield point, lb. per sq. in.	28,000 to 36,000	40,000 to 60,000[1]
Reduction of area, per cent	65–55	55–45
Elongation in 2 in., per cent	40–30	Unimportant

This steel will case-harden but is not as suitable for this purpose as Steel No. 1020, a note on which follows.

[1] *These high yield points can be obtained only in comparatively light or small sections, either in the sheet or rod form; say ½-in. round or ¼-in. sheets or flats.*

Specification No. 1020

0.20 Carbon

This steel is known to the trade as 0.20 carbon, open-hearth steel, and often as machine steel.

This quality is intended primarily for case-hardening. It forges well and machines well, but should not be considered as screw machine stock. It may therefore be used for a very large variety of forged, machined and case-hardened parts of an automobile where strength is not paramount.

Steel of this quality may also be drawn into tubes and rolled into cold rolled forms, and, as a matter of fact, makes a better frame than Steel No. 1010, because of the slightly higher carbon and resulting strength. The increased carbon content has no detrimental effect as far as usage is concerned, and it is only the most difficult of cold-forming operations that cause it to crack during the forming. For automobile parts it may be safely used interchangeably with Steel No. 1010 as far as cold pressed shapes are concerned.

Heat treatment of this steel produces but little change as far as strength is concerned, but does cause a desirable refinement of grain after forging, and materially increases the toughness, facts that are plainly shown under "Physical Characteristics," page 264. The following treatment, which will often help the machining qualities, is all that is necessary:

Heat Treatment H

After forging or machining—

1. Heat to 1500 to 1600°F.
2. Quench.
3. Reheat to 600 to 1200°F. and cool slowly.

Case-hardening is the most important treatment for this quality of steel. The character of the operation must depend upon the importance of the part to be treated and upon the shape and size. There is a certain group of parts in an automobile which are not called upon to carry much load or withstand any shock. The principal requirement is hardness. Such parts are fairly illustrated by screws and by rod-end pins. The simplest form of case-hardening will suffice, as follows:

Heat Treatment A

After forging or machining—

1. Carbonize at a temperature between 1600 and 1750°F. (1650 to 1700°F. desired).
2. Cool slowly or quench.
3. Reheat to 1450 to 1500°F. and quench.

Another class of parts demands the best treatment (Heat Treatment B), such as gears, steering-wheel pivot-pins, cam-rollers, push-rods and many similar details of an automobile which the manufacturer learns by experience must not only be hard on the exterior surface but must possess strength as well. The desired treatment is one which first refines and strengthens the interior and uncarbonized metal. This is then followed by a treatment which refines the exterior, carbonized, or high carbon metal.

Heat Treatment B

After forging or machining—

1. Carbonize at a temperature between 1600 and 1750°F. (1650 to 1700°F. desired).
2. Cool slowly in the carbonizing mixture.
3. Reheat to 1550 to 1625°F.
4. Quench.
5. Reheat to 1400 to 1450°F.
6. Quench.
7. Draw in hot oil at a temperature which may vary from 300 to 450 °F., depending upon the degrees of hardness desired.

In the case of very important parts, the last drawing operation should be continued from one to three hours, to insure the full benefit of the operation.

The objects of drawing are twofold: First, and not least important, is the relieving of all internal strains produced by quenching; second is the decrease in hardness, which is sometimes desirable. The hardness begins to decrease very materially from 350°F. up, and the operation must be controlled as dictated by experience with any given part.

There are certain very important pieces that demand all of these operations, but the last drawing operation can be omitted with a large number. Experience teaches what degree of hardness and toughness combined is necessary for any given part. It is impossible to lay down a general rule covering all different uses. If the fundamental principle is well understood, there should be no trouble in developing the treatment to a proper degree.

Following the foregoing treatment, a fractured part should show a fine-grained exterior, without any appearance of shiny crystals. The smaller the crystals the better. The interior may show a silky, fibrous condition or a fine crystalline condition; but it must not show a coarse, shiny, crystalline condition.

When cold rolled or cold drawn, this steel will have a yield point of 40,000 to 75,000[1] lb. per sq. in., and a reduction of area from 35 to 30 per cent.

[1]*In sections not over ½ in. round or ¼-in. sheets or flats.*

PHYSICAL CHARACTERISTICS OF HEAT-TREATED S. A. E. STEEL. NO. 1020

Heat Treatment

The accompanying data apply to ½-in. to 1½-in. round specimens which were heated from 15 to 30 min. at 1560 to 1580°F.; quenched in oil; reheated for 30 min. at temperatures indicated, and finally cooled in air. (Heat Treatment H.)

TABLE XXI

Reheating temperature, °F.	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Reduction of area, per cent.	Elongation in 2 in., per cent.	Brinell hardness	Scleroscope hardness
400	80,000	50,000	60.0	20.0	180	34
500	79,000	49,000	60.5	20.5	175	34
600	78,000	48,000	61.0	21.0	170	34
700	77,000	46,500	62.0	22.5	160	33
800	76,000	44,500	63.5	24.5	150	33
900	75,000	42,500	65.0	26.5	140	32
1000	74,000	40,500	66.5	28.5	130	32
1100	73,000	38,500	68.0	30.0	120	31
1200	72,000	37,000	69.0	31.5	110	31
1300	71,000	36,000	69.5	32.0	105	30
1400	70,000	35,000	70.0	32.5	100	30

[1] Values are average minimum, except those for hardness, which are average.

Specification No. 1025

0.25 *Carbon*

This steel is used most widely for frames and for ordinary drop forgings where moderate ductility is desired, but high strength is not essential. Heat treatment has a moderate effect on its physical properties but this effect is not nearly so marked as on Steel No. 1035.

Heat Treatment H or D may be used for this quality of steel.

Heat Treatment H is the simplest form of heat treatment; the drawing operation (No. 3) must be varied to suit each individual case. If great toughness and little increase in strength are desired, the higher drawing temperatures may be used, that is in the neighborhood of 1100 to 1200 °F. If much strength is desired and little toughness, the lower temperatures are available. Even the lowest of the temperatures given will produce a quality of steel, after oil quenching, that is very tough—sufficiently tough for many important parts. In fact, with some parts the drawing operation (No. 3) can be entirely omitted.

Results better than obtainable with the above sequence of operations can be obtained by a double treatment as follows:

Heat Treatment D

After forging or machining—

1. Heat to 1500 to 1600°F.
2. Quench.
3. Reheat to 1450 to 1500°F.
4. Quench.
5. Reheat to 600 to 1200°F. and cool slowly.

This produces a refinement of grain not possible with one treatment and is resorted to in parts where extremely good qualities are desired.

This quality of steel is not intended for case-hardening, but by careful manipulation it may be so treated. This should be done in emergencies only, rather than as a regular practice and, if at all, only with the double treatment followed by the drawing operation; that is, with the most painstaking form of case-hardening.

PHYSICAL CHARACTERISTICS OF HEAT-TREATED S. A. E. STEEL NO. 1025

Heat Treatment

The accompanying data apply to ½-in. to 1½-in. round specimens which were heated from 15 to 30 min. at 1540 to 1560°F.; quenched in oil; reheated for 30 min. at temperatures indicated, and finally cooled in air. (Heat Treatment H.)

TABLE XXII[1]

Reheating temperature, °F.	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Reduction of area, per cent.	Elongation in 2 in., per cent.	Brinell hardness	Scleroscope hardness
400	90,000	60,000	55.0	17.0	215	37
500	89,000	59,000	55.5	17.5	210	37
600	88,000	57,000	56.0	18.5	200	36
700	86,500	55,000	57.5	20.0	185	36
800	84,500	52,500	59.0	21.5	175	35
900	82,500	50,000	61.0	23.5	160	34
1000	80,500	47,500	63.0	25.5	145	33
1100	78,500	45,000	64.5	27.0	135	32
1200	77,000	43,000	66.0	28.5	125	31
1300	76,000	41,000	67.0	29.5	115	30
1400	75,000	40,000	67.5	30.0	110	30

[1] Values are average minimum, except those for hardness, which are average.

SPECIFICATION No. 1035

0.35 *Carbon*

This material is sometimes referred to in the trade as 0.35 carbon machine steel.

It is primarily for use as a structural steel. It forges well, machines well and responds to heat treatment in the matter of strength as well as toughness; that it is to say, intelligent treatment will produce marked increase in the yield point. It can be used for all forgings, such as axles, driving shafts, steering pivots and other structural parts. It is the best all-around structural steel for such use as its strength warrants.

Heat treatment for toughening and strength is of importance with this steel. The heat treatment must be modified in accordance with the experience of the individual user, to suit the size of the part treated and the combination of strength and toughness desired. The steel should be heat-treated in all cases where reliability is important.

Machining may precede the heat treatment given below, depending somewhat upon convenience and the character of the treatment. If the highest strength is demanded, a strong quenching medium must be employed; for example, brine. In such case the yield point will be correspondingly high and the steel correspondingly hard and difficult to machine. On the other hand, if a moderately high yield point is all that is desired, an oil quench will suffice and machining may follow without any difficulty whatever.

Heat Treatment H, D or E may be used on this quality of steel. When Heat Treatment E is applied, machining may follow operation 2.

Heat Treatment E

After forging or machining—

1. Heat to 1500 to 1550°F.

2. Cool slowly.
3. Reheat to 1450 to 1500°F.
4. Quench.
5. Reheat to 600 to 1200°F. and cool slowly.

PHYSICAL CHARACTERISTICS OF HEAT-TREATED S. A. E. STEEL NO. 1035

Heat Treatment

The accompanying data apply to ½-in. to 1½-in. round specimens which were heated from 15 to 30 min. at 1510 to 1530° F.; quenched in oil; reheated for 30 min. at temperatures indicated; and finally cooled in air. (Heat Treatment H.)

Reheating temperature, °F.	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Reduction of area, per cent.	Elongation in 2 in., per cent.	Brinell hardness	Scleroscope hardness
400	105,000	75,000	42.5	15.0	260	42
500	104,000	74,000	43.5	15.5	255	42
600	102,500	72,000	45.0	16.5	245	41
700	100,000	69,000	47.0	18.0	235	40
800	97,000	66,000	49.5	19.5	220	39
900	94,000	63,000	52.5	21.5	200	37
1000	91,000	59,500	55.5	23.5	180	35
1100	88,000	56,000	58.0	25.0	165	34
1200	85,500	53,000	60.0	26.5	150	33
1300	83,500	51,000	61.5	27.5	140	32
1400	82,000	50,000	62.5	28.0	135	32

SPECIFICATION No. 1045

0.45 *Carbon*

This material is ordinarily known to the trade as 0.45 carbon machine steel. This quality represents a structural steel of greater strength than Steel No. 1035. Its uses are more limited and are confined in a general way to such parts as demand a high degree of strength and a considerable degree of toughness. At the same time with proper heat treatment the fatigue-resisting (endurance) qualities are very high—higher than those of any of the foregoing steels.

This steel is commonly used for crankshafts, driving shafts and propeller shafts. It has also been used for transmission gears, but it is not quite hard enough without case-hardening and is not tough enough with case-hardening to make safe transmission gears. It should not be used for case-hardened parts. Other specifications are decidedly better for this purpose.

In a properly annealed condition it machines well—not well enough for screw machine work, but certainly well enough for all-round machine shop practice. Heat Treatment E provides the annealing operation when needed, machining to follow operation 2; this treatment is especially adapted

to crankshafts and similar parts. Heat Treatment H is also commonly used for this quality of steel.

PHYSICAL CHARACTERISTICS OF HEAT-TREATED S. A. E. STEEL NO. 1045

Heat Treatment

The accompanying data apply to ½-in. to 1½-in. round specimens which were heated from 15 to 30 min. at 1490 to 1510°F.; quenched in oil; reheated for 30 min. at temperatures indicated; and finally cooled in air. (Heat Treatment H.)

TABLE XXIII

Reheating temperature, °F.	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Reduction of area, per cent.	Elongation in 2 in., per cent.	Brinell hardness	Scleroscope hardness
400	125,000	90,000	35.0	12.5	300	45
500	123,500	88,000	36.0	13.0	290	45
600	121,000	85,500	37.0	13.5	280	44
700	118,000	82,500	39.0	14.5	265	43
800	114,000	79,000	42.0	16.0	250	41
900	110,000	75,000	45.0	17.5	230	40
1000	106,000	71,000	48.0	19.0	210	38
1100	102,000	67,000	50.5	20.5	195	37
1200	99,000	63,500	53.0	21.5	180	36
1300	96,500	61,500	54.0	22.0	165	35
1400	95,000	60,000	55.0	22.5	160	35

SPECIFICATION No. 1095

0.95 *Carbon*

This is a grade of steel used generally for springs. Properly heat treated, extremely good results are possible.

The hardening and drawing of springs, that is, the heat treatment of them, is, as a rule, in the hands of the springmaker, but in case it is desired to treat, as for small springs, the following is recommended:

Heat Treatment F

After shaping or coiling—

1. Heat to 1425 to 1475°F.
2. Quench in oil.
3. Reheat to 400 to 900°F in accordance with degree of temper desired, and cool slowly.

It must be understood that the higher the drawing temperature (Operation 3), the lower will be the yield point of the material. On the other hand, if the material be drawn at too low a temperature, it will be brittle. A few practical trials will locate the best temper for any given shape or size.

The physical characteristics of heat-treated spring steel are best determined by transverse test. This is because steel as hard as tempered spring steel is very difficult to hold firmly in the jaws of a tensile testing machine. There is more or less slip, and side strains are bound to occur, all of which tends to produce misleading results.

The physical characteristics in the annealed condition may be omitted, inasmuch as this grade of steel is not ordinarily used for structural parts in such condition.

Careful examination of the fracture of the treated material is desirable. After tempering, no suitable spring steel should be coarsely crystalline. It should be finely crystalline, and in some cases will show a partly fibrous fracture.

Physical Characteristics

(Transverse Test)

	Heat Treatment F
Elastic limit (initial set) lb. per sq. in.	90,000 to 180,000

Reduction of area and elongation are not determined in a transverse test.

SCREW STOCK

SPECIFICATION No. 1114

0.08 *to* 0.20 *Carbon*

This steel may be made by any process. It is intended for use where high screw-machine production is the important factor, strength and toughness being secondary considerations. Its composition and texture are of such nature as to permit the rapid removal of metal and a resulting smoothness of finish.

STEEL CASTINGS

SPECIFICATION No. 1235

(*Carbon as Required for Physical Properties*)

In the following remarks, genuine steel castings, and not malleable iron or complex mixtures often found in the market masquerading under the name of steel, are referred to.

All steel castings should be annealed and some shapes may be heat-treated to great advantage. Like other castings, steel castings are subject to blowholes. Consequently, they should not be used in the vital parts of an automobile. It is impossible to inspect against blow-holes, and steel castings for axles, crankshafts and steering spindles are used only at great risk. Freedom from blowholes and proper physical condition are of more importance than the absolute analysis.

On account of the great influence of varying types of foundry practice upon the properties of castings, it has not been found feasible to give a closer specification for chemical composition than that quoted under No. 1235. If it is desired to buy steel castings under precise specifications, the following, based upon the "Specifications for Steel Castings, Class B, Serial Designation A 27-14," of the American Society for Testing Materials, can be used;

I. *Manufacture*

1. The steel may be made by any process approved by the purchaser. Three grades are recognized: hard, medium, and soft.

2. All castings shall be allowed to become cold; they shall then be reheated uniformly to the proper temperature to refine the grain, and allowed to cool uniformly and slowly.

II. *Chemical Properties and Tests*

3. No casting, on check analysis, shall show over .05 per cent. phosphorus or sulphur. The carbon content shall be suitable for the physical tests and service required.

4. Drillings for analysis shall be so taken as to represent the average composition of the casting.

III. *Physical Properties and Tests*

5. The finished castings shall conform to the following minimum requirements as to tensile properties:

	Hard	Medium	Soft
Tensile strength, lb. per sq. in.	80,000	70,000	60,000
Yield point, lb. per sq. in.	36,000	31,500	27,000
Reduction of area, per cent	20	25	30
Elongation in 2 in., per cent	15	18	22

6. The test-specimen for soft castings shall bend cold through 120°, and for medium castings through 90°, around a 1-in. pin without cracking on the outside. Hard castings shall not be subject to bend-test requirements.

7. In the case of small or unimportant castings, a test to destruction on three castings from a lot may be substituted for the tension and bend tests. This test shall show the material to be ductile free from injurious defects, and suitable for the purpose intended. A lot shall consist of all castings from one melt, in the same annealing charge. In case test bars are cast separate, they shall be annealed with the lot they represent, the method of casting such test bars, or of casting test bars attached to castings to be agreed upon by purchaser and manufacturer.

8. Tension test-specimens shall be machined to the standard S. A. E. form; bend test-specimens shall be machined to 1 by ½ in. in section, with corners rounded to a radius not over 1/16 in.

9. One tension and one bend test shall be made from each annealing charge. If more than one melt is represented in an annealing charge, one tension and one bend test shall be made from each melt.

10. If any test-specimen shows defective machining or develops flaws, it may be discarded; in which case another specimen may be selected by the manufacturer and the purchaser.

11. A retest shall be allowed if the percentage of elongation is less than that specified, or if any part of the fracture is more than ¾ in. from the center of the gauge length, as indicated by scribe scratches marked on the specimen before testing.

IV. *Workmanship and Finish*

12. The finished castings shall conform substantially to the sizes and shapes of the patterns, shall be made in a workmanlike manner, and be free from injurious defects.

13. Minor defects which do not impair the strength of the castings may, with the approval of the purchaser, be welded by an approved process. The defects shall first be cleaned out to solid metal; and after welding the castings shall be annealed.

14. Castings offered for inspection shall not be painted or covered with any substance that will hide defects, nor rusted to such an extent as to hide defects.

ALLOY STEELS

Remarks on Use

In connection with the purchase and use of alloy steels it should be borne in mind that such steels should be used in the treated condition only, that is, not in an annealed or natural condition. In the latter condition there is a slight benefit, perhaps, as compared with plain carbon steels, but as a rule nothing commensurate with the increased cost. In the heat-treated condition, however, there is a very marked improvement in physical characteristics.

NICKEL STEELS

Specification No. 2315

0.15 Carbon, 3½ per cent. Nickel

This quality of steel is embraced in these specifications to furnish a nickel steel that is suitable for carbonizing purposes. Steel of this character, properly carbonized and heat treated, will produce a part with an exceedingly tough and strong core, coupled with the desired high carbon exterior.

This steel is also available for structural purposes, but is not one to be selected for such purpose when ordering materials. Much better results will be obtained with one of the other nickel steels of higher carbon.

It is intended for casehardened gears, for both the bevel driving and transmission systems, and for such other casehardened parts as demand a very tough, strong steel with a hardened exterior.

The casehardening sequence may be varied considerably, as with Steel No. 1020, those parts of relatively small importance requiring a simpler form of treatment. As a rule, however, those parts which require the use of nickel steel require the best type of casehardening, as follows·

Heat Treatment G

After forging or machining—

1. Carbonize at a temperature between 1600 and 1750°F. (1650 to 1700°F. desired).
2. Cool slowly in the carbonizing material.

3 Reheat to 1500 to 1550°F.
4. Quench.
5. Reheat to 1300 to 1400°F.
6. Quench.
7. Reheat to 250 to 500°F. (in accordance with the necessities of the case) and cool slowly.

The second quench (Operation 6) must be conducted at the lowest possible temperature at which the material will harden. It will be found that sometimes this is lower than 1300°F.

In connection with certain uses it will be found possible to omit the final drawing (Operation 7) entirely, but for parts of the highest importance this operation should be followed as a safeguard. Parts of intricate shape, with sudden changes of thickness, sharp corners and the like, particularly sliding gears, should always be drawn, in order to relieve the internal strains.

Much is to be learned from the character of the fracture. The center should be fibrous in appearance, and the exterior, high-carbon metal closely crystalline, or even silky.

When used for structural purposes, the physical characteristics will range about as follows:

Physical Characteristics

	Annealed	Heat Treatment H or K
Yield point, lb. per sq. in.	35,000 to 45,000	40,000 to 80,000
Reduction of area, per cent	65–45	65–40
Elongation in 2 in., per cent	35–25	35–15

Specification No. 2320

0.20 *Carbon, 3½ per cent. Nickel*

This quality may be used interchangeably with Steel No. 2315. Although intended primarily for casehardening, it can be properly used for structural parts, with suitable heat treatment, and will give elastic limits somewhat higher than material provided by the preceding specification.

For casehardening, Heat Treatment G should be followed, and for structural purposes the treatment should be in accordance with Heat Treatment H or K; the quenching temperatures, as with other steels, being modified to meet individual cases.

Heat Treatment K

After forging or machining—

1. Heat to 1500 to 1550°F.
2. Quench.
3. Reheat to 1300 to 1400°F.
4. Quench.
5. Reheat to 600 to 1200°F. and cool slowly.

Physical Characteristics

	Annealed	Heat Treatment H or K
Yield point, lb. per sq. in.	40,000 to 50,000	50,000 to 125,000
Reduction of area, per cent	65–40	65–40
Elongation in 2 in., per cent	30–20	25–10

PHYSICAL CHARACTERISTICS OF HEAT-TREATED S. A. E. STEEL NO. 2320

Heat Treatment

The accompanying data apply to ½-in. to 1½-in. round specimens, heated from 15 to 30 min. at 1510 to 1540°F.; quenched in oil; reheated for 30 min. at temperatures indicated and finally cooled in air. (Heat Treatment H.)

TABLE XXIV[1]

Reheating temperature, °F.	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Reduction of area, per cent.	Elongation in 2 in., per cent.	Brinell hardness	Scleroscope hardness
400	170,000	140,000	45.0	11.0	375	55
500	168,000	136,500	46.0	12.0	368	54
600	162,000	130,000	48.5	13.5	355	52
700	155,000	123,000	51.5	15.5	340	50
800	145,000	112,000	55.5	18.5	310	46
900	130,000	99,000	60.5	21.5	280	42
1000	112,000	84,000	65.5	25.0	240	38
1100	96,000	68,000	69.5	27.0	200	34
1200	82,000	54,000	72.5	29.0	164	31
1300	75,000	45,000	74.5	29.5	140	29
1400	70,000	40,000	75.0	30.0	125	28

[1] Values are average minimum, except those for hardness, which are average.

SPECIFICATION NO. 2330

0.30 *Carbon,* 3½ *per cent. Nickel*

This quality of steel is primarily for heat-treated structural parts where strength and toughness are sought; such parts as axles, front-wheel spindles, crankshafts, driving shafts and transmission shafts. Wide variations of yield point or elastic limit are possible by the use of different quenching mediums—oil, water or brine—and variation in drawing temperatures, from 500 up to 1200°F. (Heat Treatment H.)

A higher refinement of this treatment is Heat Treatment K.

Physical Characteristics

The physical characteristics of this steel may be considered as practically those obtained with Steel No. 2320, slight modifications in the treatment much more than offsetting the slight difference in the carbon content.

	Annealed	Heat Treatment H or K
Yield point or elastic limit, lb. per sq. in.	40,000 to 50,000	60,000 to 130,000
Reduction of area, per cent	60–40	60–30
Elongation in 2 in., per cent	30–20	25–10

PHYSICAL CHARACTERISTICS OF HEAT-TREATED S. A. E. STEEL NO. 2330

Heat Treatment

The accompanying data apply to ½-in. to 1½-in. round specimens, heated from 15 to 30 min. at 1485 to 1515°F.; quenched in oil; reheated for 30 min. at temperatures indicated and finally cooled in air. (Heat Treatment H or M.)

TABLE XXV[1]

Reheating temperature, °F.	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Reduction of area, per cent.	Elongation in 2 in., per cent.	Brinell hardness	Scleroscope hardness
400	220,000	190,000	35.0	10.0	436	60
500	210,000	182,000	37.0	10.7	420	59
600	198,000	170,000	40.0	11.5	400	57
700	180,000	154,000	44.0	12.7	375	54
800	160,000	135,000	49.0	14.5	340	50
900	140,000	115,000	54.0	16.0	300	46
1000	120,000	95,000	59.0	18.0	250	41
1100	104,000	77,000	62.7	20.5	210	37
1200	92,000	64,000	66.0	22.5	180	34
1300	85,000	55,000	68.5	24.5	162	32
1400	80,000	50,000	70.0	25.0	150	30

[1] Values are average minimum, except those for hardness, which are average.

SPECIFICATION NO. 2335

0.35 *Carbon, 3½ per cent. Nickel*

This quality of steel is subject to precisely the same remarks as Steel No. 2330. It will respond a little more sharply to heat treatment and can be forced to higher elastic limits. The difference will be small except in extreme cases.

Physical Characteristics

	Annealed	Heat Treatment H or K
Yield point or elastic limit, lb. per sq. in.	45,000 to 55,000	65,000 to 160,000
Reduction of area, per cent	55–35	55–25
Elongation in 3 in., per cent	25–15	25–10

Specification No. 2340

0.40 *Carbon,* 3½ *per cent. Nickel*

The above nickel steel is a quality not in wide use but available for certain purposes. The carbon content being higher than generally used, greater hardness is obtainable by quenching; and as increased brittleness accompanies the greater hardness, the treatments given must be modified to meet such condition. For example, the final quench may be at a considerably lower temperature, and the final drawing temperature, or partial annealing, must be carefully chosen, in order to produce the desired toughness and other physical characteristics.

Physical Characteristics

	Annealed	Heat Treatment H or K
Yield point or elastic limit, lb. per sq. in.	55,000 to 65,000	70,000 to 200,000
Reduction of area, per cent..........	50–30	55–15
Elongation in 2 in., per cent.........	25–15	25–5

PHYSICAL CHARACTERISTICS OF HEAT-TREATED S. A. E. STEEL NO. 2340

Heat Treatment

The accompanying data apply to ½-in. to 1½-in. round specimens, heated from 15 to 30 min. at 1435 to 1465°F.; quenched in oil; reheated for 30 min. at temperatures indicated by the abscissas of the curves; and finally cooled in air. (Heat Treatment M.)

Table XXVI[1]

Reheating temperature, °F.	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Reduction of area, per cent.	Elongation in 2 in., per cent.	Brinell hardness	Scleroscope hardness
400	240,000	215,000	32.5	10.0	450	70
500	230,000	204,000	24.5	11.0	427	65
600	215,000	190,000	37.5	12.0	400	61
700	196,000	171,000	42.0	13.0	370	56
800	175,000	150,000	47.0	14.0	335	51
900	155,000	130,000	51.0	16.0	295	46
1000	135,000	110,000	55.0	18.0	260	42
1100	117,000	92,000	58.0	20.0	235	38
1200	105,000	78,000	60.0	21.5	215	36
1300	96,000	69,000	61.0	22.0	205	35
1400	90,000	60,000	62.5	22.5	200	35

[1] Values are average minimum, except those for hardness, which are average.

NICKEL CHROMIUM STEELS

Remarks on Use

In general it can be said in the case of the nickel chromium steels that the heat treatments and the properties induced thereby are much the same as in the case of plain nickel steels, except that the effects of the heat treatments are somewhat augmented by the presence of the chromium, and further that these effects increase with increasing amounts of nickel and chromium.

Specification No. 3120

0.20 *Carbon*

This quality of steel is intended primarily for case-hardening (Heat Treatment G). It may also be used for structural parts with suitable heat treatment (Heat Treatment H or D). It should not be used in the natural or untreated condition.

Physical Characteristics

	Annealed	Heat Treatment H or D
Yield point or elastic limit, lb. per sq. in.	30,000 to 40,000	40,000 to 100,000
Reduction of area, per cent..........	55–40	65–40
Elongation in 2 in., per cent.........	35–25	25–15

PHYSICAL CHARACTERISTICS OF HEAT-TREATED S. A. E. STEEL NO. 3120

Heat Treatment

The accompanying data supply to ½-in. to 1½-in. round specimens, heated from 15 to 30 min. at 1585 to 1615°F.; quenched in oil; reheated for 30 min. at temperatures indicated and finally cooled in air. (Heat Treatment H or M.)

Table XXVII

Reheating temperature, °F.	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Reduction of area, per cent.	Elongation in 2 in., per cent.	Brinell hardness	Scleroscope hardness
400	160,000	120,000	52.5	15.0	275	....
500	155,000	116,000	54.0	15.5	265	....
600	148,000	110,000	57.0	16.0	250	....
700	137,000	102,000	61.0	16.5	240	....
800	125,000	95,000	65.5	18.0	225	....
900	111,000	84,000	69.0	21.0	205	....
1000	100,000	74,000	71.0	24.5	185	....
1100	91,000	66,000	71.5	28.5	175	....
1200	84,000	60,000	72.0	31.5	160	....
1300	80,000	54,000	72.5	33.5	150	....
1400	75,000	50,000	72.5	35.0	150	....

SPECIFICATION NOS.

3125......0.25 *Carbon*
3130......0.30 *Carbon*
3135......0.35 *Carbon*
3140......0.40 *Carbon*

These qualities of steel are intended primarily for structural purposes in a heat-treated condition (Heat Treatment H, D or E). Steel No. 3125 may be used for case-hardening, as also may Steel No. 3130 if necessary.

Physical Characteristics

Steels Nos. 3125, 3130:	Annealed	Heat Treatment H, D or E
Yield point or elastic limit, lb. per sq. in	40,000 to 55,000	50,000 to 125,000
Reduction of area, per cent..........	50–35	55–25
Elongation in 2 in., per cent.........	30–20	25–10

PHYSICAL CHARACTERISTICS OF HEAT-TREATED S. A. E. STEEL No. 3130.

Heat Treatment

The accompanying data apply to ½ to 1½-in. round specimens, heated from 15 to 30 min. at 1535 to 1565°F.; quenched in oil; reheated for 30 min. at temperatures indicated and finally cooled in air. (Heat Treatment H.)

TABLE XXVIII

Reheating temperature, °F.	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Reduction of area, per cent.	Elongation in 2 in., per cent.	Brinell hardness	Scleroscope hardness
400	190,000	155,000	37.5	10.0	....	50
500	188,000	150,000	41.0	11.0	....	49
600	180,000	140,000	46.0	12.5	....	48
700	167,000	128,000	52.5	13.5	....	46.
800	150,000	115,000	59.0	15.5	....	43
900	134,000	102,000	63.0	17.5	....	40
1000	120,000	90,000	65.0	20.0	....	38
1100	104,000	81,000	66.5	23.5	....	35
1200	92,000	76,000	68.0	26.5	....	32
1300	86,000	72,000	69.0	28.5	....	31
1400	80,000	70,000	70.0	30.0	....	30

Steels 3135, 3140:	Annealed	Heat Treatment H, D or E
Yield point or elastic limit, lb. per sq. in.	45,000 to 60,000	55,000 to 150,000
Reduction of area, per cent...........	45–30	50–25
Elongation in 2 in., per cent..........	25–15	20– 5

PHYSICAL CHARACTERISTICS OF HEAT-TREATED S. A. E. STEEL NO. 3140

Heat Treatment

The accompanying data apply to ½ to 1½-in. round specimens, heated from 15 to 30 min. at 1485 to 1515°F; quenched in oil; reheated for 30 min. at temperatures indicated and finally cooled in air. (Heat Treatment H or M.)

TABLE XXIX

Reheating temperature °F	Tensile strength, lb. per sq. in.	Elastic limit, lb. per sq. in.	Reduction of area, per cent.	Elongation in 2 in., per cent.	Brinell hardness	Scleroscope hardness
400	230,000	200,000	27.0	7.5	425	65
500	226,000	195,000	28.0	8.0	410	64
600	220,000	185,000	30.0	9.0	390	62
700	204,000	168,000	34.0	10.5	360	59
800	182,000	148,000	39.0	12.5	330	56
900	157,000	126,000	46.5	14.0	300	52
1000	130,000	105,000	52.5	16.0	275	47
1100	112,000	94,000	56.5	17.0	245	42
1200	100,000	84,000	60.0	18.0	225	38
1300	93,000	80,000	61.0	19.0	215	36
1400	90,000	75,000	62.0	20.0	210	35

SPECIFICATION NO. 3220

0.20 *Carbon*

This steel is intended for case-hardened parts of nickel chromium steel. Case-hardened parts demanding this grade of steel also demand the most careful heat treatment (Heat Treatment G). It may also be used for structural purposes with Heat Treatment H or D.

Physical Characteristics

	Annealed	Heat Treatment H or D
Yield point or elastic limit, lb. per sq. in.	35,000 to 45,000	45,000 to 120,000
Reduction of area, per cent..........	60–45	65–30
Elongation in 2 in., per cent.........	25–20	20– 5

SPECIFICATION NO. 3230

0.30 *Carbon*

This steel is intended for the most important structural parts and should be used only in a heat treated condition (Heat Treatment H or D).

Physical Characteristics

	Annealed	Heat Treatment H or D
Yield point or elastic limit, lb. per sq. in.	40,000 to 50,000	60,000 to 175,000
Reduction of area, per cent..........	55–40	60–30
Elongation in 2 in., per cent.........	25–15	20– 5

Specification No. 3240

0.40 *Carbon*

This quality of steel is suitable for structural parts where unusual strength is demanded. Higher elastic limit is possible under a given treatment than with material like steel No. 3230. The toughness will not be quite as great, but this does not bar the material from uses where toughness is not the controlling factor and where strength is.

Heat Treatment H or D is recommended.

Physical Characteristics

	Annealed	Heat Treatment H or D
Yield point or elastic limit, lb. per sq. in.	4.000 to 60,000	65,000 to 200,000
Reduction of area, per cent..........	50–40	50–20
Elongation in 2 in., per cent.........	25–15	15– 2

Specification No. 3250

0.50 *Carbon*

This steel is intended for gears where extreme strength and hardness are necessary.

To heat treat for gears either Heat Treatment M or Q should be followed, the latter giving the better results.

Heat Treatment M

After forging or machining—

1. Heat to 1450 to 150°F.
2. Quench.
3. Reheat to 500 to 1250°F. and cool slowly.

A higher refinement of this same treatment is:

Heat Treatment Q

After forging—

1. Heat to 1475 to 1525°F. (Hold at this temperature one-half hour to insure thorough heating.)
2. Cool slowly.
3. Machine.
4. Reheat to 1375 to 1425°F.
5. Quench.
6. Reheat to 250 to 550°F. and cool slowly.

Physical Characteristics

	Annealed	Heat Treatment M or Q
Yield point or elastic limit, lb. per sq. in.	50,000 to 60,000	150,000 to 250,000
Reduction of area, per cent..........	50–40	25–15
Elongation in 2 in., per cent.........	25–15	15– 2

Specification No. X3315

0.15 Carbon

This steel is intended primarily for case-hardening. It is higher in nickel and chromium than the preceding nickel chromium steels. Heat Treatment G should be followed.

It is sometimes used for structural parts, when Heat Treatment M is applicable.

Physical Characteristics

	Annealed	Heat Treatment M
Yield point or elastic limit, lb. per sq. in.	35,000 to 45,000	40,000 to 100,000
Reduction of area, per cent..........	60–45	65–30
Elongation in 2 in., per cent.........	25–20	20– 5

Specification No. X3335

0.35 Carbon

This steel is intended for structural parts of the most important character, such as crankshafts, axles, spindles, drive shafts and transmission shafts. Heat Treatment P or R is recommended.

This steel is not intended for case-hardening.

Heat Treatment P

After forging or machining—

1. Heat to 1450 to 1500°F.
2. Quench.
3. Reheat to 1375 to 1450°F.
4. Quench.
5. Reheat to 500 to 1250°F. and cool slowly.

Heat Treatment R

After forging—

1. Heat to 1500 to 1550°F.
2. Quench in oil
3. Reheat to 1200 to 1300°F. (Hold at this temperature three hours.)
4. Cool slowly.
5. Machine.
6. Reheat to 1350 to 1450°F.

7. Quench in oil.
8. Reheat to 250 to 500°F. and cool slowly.

Physical Characteristics

	Annealed	Heat Treatment P or R
Yield point or elastic limit, lb. per sq. in.	45,000 to 55,000	60,000 to 175,000
Reduction of area, per cent..........	55–40	60–30
Elongation in 2 in., per cent.........	25–15	20– 5

Specification No. X3350

0.50 *Carbon*

This steel is an alternative quality for gears. The remarks made on Steel No. 3250 apply to this case. The physical characteristics are similar to those of Steel No. 3250. Heat Treatment R should be used, although P is applicable.

Specification No. 3320

0.20 *Carbon*

The remarks made in connection with Steel No. 3220 apply to this steel also. There is no appreciable difference in the physical characteristics. Carbonizing should follow the practice indicated under Heat Treatment L.

Heat Treatment L

After forging or machining—

1. Carbonize at a temperature between 1600 and 1750°F. (1650 to 1700°F. desired).
2. Cool slowly in the carbonizing mixture.
3. Reheat to 1400 to 1500°F.
4. Quench.
5. Reheat to 1300 to 1400°F.
6. Quench.
7. Reheat to 250 to 500°F. and cool slowly.

Specification No. 3330

0.30 *Carbon*

This steel, like No. 3230, is intended for very important structural parts. The high nickel and chromium contents make it exceedingly tough and strong when treated according to Heat Treatments P or R.

Specification No. 3340

0.40 *Carbon*

This steel is suitable for gears to be hardened without carbonizing. The remarks made in connection with Steels Nos. 3240 and 3250 apply. Heat Treatments P or R should be used.

CHROMIUM STEELS

Specification No. 5120

0.20 Carbon

This steel is similar in properties to Nos. 2320 and 3120 in that it is a case-hardening grade of much better quality than carbon steel. Heat Treatment B should be used.

Specification No. 5140

0.40 Carbon

This grade of steel is very similar in properties to Steel No. 3140. When treated according to H or D it becomes useful for high-duty shafting, etc. The drawing temperature should be moderately high in order to maintain a safe degree of toughness.

Specifications Nos.

5195......0.95 *Carbon*
51120......1.20 *Carbon*
5295......0.95 *Carbon*
52120......1.20 *Carbon*

These four grades of steel are used almost exclusively for ball bearing cups and cones, where their extreme hardness is indispensable. The treatment of these steels is in the hands of specialists, but in a general way Treatments P and R illustrate the procedures followed.

CHROMIUM VANADIUM STEELS

Specification No. 6120

0.20 Carbon

This quality is primarily for case-hardening. It is used for the most important case-hardened parts; that is, case-hardened shafts, gears and the like.

This steel may also be used in a heat treated condition for structural purposes, but for such work some of the steels following are to be preferred, particularly where higher strength is desired.

The case-hardening treatment recommended is that covered by Heat Treatment S.

Heat Treatment S

After forging or machining—

1. Carbonize at a temperature between 1600 and 1750°F. (1650 to 1700°F. desired).
2. Cool slowly in the carbonizing mixture.
3. Reheat to 1650 to 1750°F.
4. Quench.
5. Reheat to 1475 to 1550°F.
6. Quench.
7. Reheat to 250 to 550°F. and cool slowly.

For structural purposes the following heat treatment is recommended:

Heat Treatment T

After forging or machining—

1. Heat to 1750 to 1700°F.
2. Quench.
3. Reheat to 500 to 1300°F. and cool slowly.

Physical Characteristics

	Annealed	Heat Treatment T
Yield point or elastic limit, lb. per sq. in.	40,000 to 50,000	55,000 to 100,000
Reduction of area, per cent..........	65–50	65–45
Elongation in 2 in., per cent.........	30–20	25–10

SPECIFICATION No. 6125

0.25 *Carbon*

The difference between this and the preceding steel is very slight and they may be used interchangeably for structural purposes. This steel may be case-hardened but is not first choice for this purpose.

The physical characteristics can be considered as practically the same as those given for Steel No. 6120.

Physical Characteristics

	Annealed	Heat Treatment T
Yield point or elastic limit, lb. per sq. in.	40,000 to 50,000	55,000 to 100,000
Reduction of area, per cent..........	65–50	65–45
Elongation in 2 in., per cent.........	32–20	25–10

SPECIFICATION No. 6130

0.30 *Carbon*

This quality of steel is intermediate in the carbon range and can be used interchangeably with steel No. 6125 for structural purposes. It should not be used for case-hardening. When subjected to Heat Treatment T it possesses a high degree of combined strength and toughness.

Physical Characteristics

	Annealed	Heat Treatment T
Yield point or elastic limit, lb. per sq. in.	45,000 to 55,000	60,000 to 150,000
Reduction of area, per cent..........	60–50	55–25
Elongation in 2 in., per cent.........	25–20	15– 5

SPECIFICATION No. 6135

0.35 *Carbon*

This specification provides a first-rate quality of steel for structural parts that are to be heat treated. The fatigue-resisting (endurance) qualities of this material are excellent.

Physical Characteristics

	Annealed	Heat Treatment T
Yield point or elastic limit, lb. per sq. in.	45,000 to 55,000	60,000 to 150,000
Reduction of area, per cent..........	60–50	55–25
Elongation in 2 in., per cent.........	25–20	15– 5

Specification No. 6140

0.40 *Carbon*

This is a very good quality of steel to be selected where a high degree of strength is desired, coupled with a good measure of toughness. Its fatigue-resisting qualities are very high, and it is a first-class material for high-duty shafts.

Heat Treatment T is recommended:

Physical Characteristics

	Annealed	Heat Treatment T
Yield point or elastic limit, lb. per sq. in.	50,000 to 60,000	65,000 to 175,000
Reduction of area, per cent..........	55–45	50–15
Elongation in 2 in., per cent.........	25–15	15– 2

Specification No. 6145

0.45 *Carbon*

This quality of steel contains sufficient carbon in combination with chromium and vanadium to harden to a considerable degree when quenched at a proper temperature, and may be used for gears and springs.

For structural parts where exceedingly high strength is desirable Heat Treatment T should be followed.

For gears this steel should be annealed after forging, and before machining, the anneal to consist of operations 1 and 2 of the following:

Heat Treatment U

After forging—

1. Heat to 1525 to 1600°F. (Hold at this temperature one-half hour to insure thorough heating.)
2. Cool slowly.
3. Machine.
4. Reheat to 1650 to 1700°F.
5. Quench.
6. Reheat to 350 to 550°F. and cool slowly.

This last drawing operation may be modified to obtain any desired hardness.

Physical Characteristics

	Annealed	Heat Treatment U
Yield point or elastic limit, lb. per sq. in.	55,000 to 65,000	150,000 to 200,000
Reduction of area, per cent..........	55–40	25–10
Elongation in 2 in., per cent.........	25–15	10– 2

Specification No. 6150

0.50 *Carbon*

Substantially the same remarks as made in regard to Steel No. 6145 apply to this steel. In this grade, however, we also find a material that is suitable for springs. With a proper sequence of heating, quenching and drawing, very high elastic limits are obtained.

For spring material Heat Treatment U is recommended, except that the last drawing (Operation 6) will be carried farther—probably from 700 to 1100°F. This final drawing temperature will have to vary with the section of material being handled, whether light spiral springs or heavy flat springs.

Physical Characteristics

	Annealed	Heat Treatment U
Yield point or elastic limit, lb. per sq. in.	60,000 to 70,000	150,000 to 225,000
Reduction of area, per cent..........	50–35	35–15
Elongation in 2 in., per cent.........	20–15	10– 2

SILICO-MANGANESE STEELS

Specifications Nos.
9250......0.50 *Carbon*
9260.......0.60 *Carbon*

These steels have been standardized by usage principally as spring steels. No. 9260 is also used to some extent for gears. Neither steel is suitable for use without heat treatment.

Both of these specifications are provided in order to meet the requirements of two groups of users: those who believe in relatively low carbon and high silicon, and those who desire higher carbon and lower silicon. When properly treated, their physical properties will not differ appreciably, though Steel No. 9250 will probably be slightly the tougher of the two. Heat Treatment V is suitable for both gears and springs.

Heat Treatment V

After forging or machining—

1. Heat to 1650 to 1750°F.
2. Quench.
3. Reheat to 400 to 1200°F. and cool slowly.

Steel No. 9260 will become harder when quenched in the same medium as Steel No. 9250. The latter, however, is more often quenched in water, while Steel No. 9260 is generally quenched in oil—a circumstance which

largely counteracts the influence of the composition. Furthermore, variation in the temperature of drawing will suffice to balance the properties closely.

The exact temperature for quenching and drawing, and the proper medium, should be determined for each case. In general, gears are drawn between 450 and 550°F. and springs between 800 and 1000°F.

Physical Characteristics

	Annealed	Heat treatment V
Yield point or elastic limit, lb. per sq. in	55,000 to 65,000	60,000 to 180,000
Reduction of area, per cent	45–30	40–10
Elongation in 2 in., per cent	25–20	20– 5

SILICO-MANGANESE STEELS

Composition in percentage

SPECIFICATION No. 9250

Carbon	0.45 to 0.55 (0.50 desired)
Manganese	0.60 to 0.80 (0.70 desired)
Phosphorus, not to exceed	0.045 *
Sulphur, not to exceed	0.045
Silicon	1.80 to 2.10 (1.95 desired)

SPECIFICATION No. 9260

Carbon	0.55 to 0.65 (0.60 desired)
Manganese	0.50 to 0.70 (0.60 desired)
Phosphorus, not to exceed	0.045 *
Sulphur, not to exceed	0.045
Silicon	1.50 to 1.80 (1.65 desired)

BABBITT METAL

S. A. E. SPECIFICATION No. 24

Tin	84.00%
Antimony	9.00%
Copper	7.00%

A variation of 1 per cent. either way will be permissible in the tin, and .5 per cent. either way will be permissible in the antimony and copper. The use of other than virgin metals is prohibited. No impurity will be permitted other than lead, and that not in excess of .25 per cent.

NOTE: This grade of babbitt is special owing to the large amount of copper contained therein. It is used for the connecting-rod linings of motor bearings, or any service where machinery designers are confronted with severe operating conditions.

* Steel made by the acid process may contain maximum .05 per cent. phosphorus.

BEARING METALS

White Brass

S. A. E. Specification No. 25

Copper	3.00 to 6.00%
Tin, not less than	65.00%
Zinc	28.00 to 30.00%

Metal with more than .25 per cent. impurities may be rejected.

NOTE: This alloy gives good results in automobile engines, but provision should be made to have it generously lubricated.

PHOSPHOR BRONZE

S. A. E. Specification No. 26

Copper	80.00%
Tin	10.00%
Lead	10.00%
Phosphorus	0.05 to 0.25%

Impurities in excess of .25 per cent. will not be permitted.

NOTE: This is a metal similar to that specified by many railroads for various purposes. It is an excellent composition where good anti-friction qualities are desired, standing up exceedingly well under heavy loads and severe usage. It should be used only against hardened steel in automobile construction.

BRASS CASTING METALS

Red Brass

S. A. E. Specification No 27

Copper	85.00%
Tin	5.00%
Lead	5.00%
Zinc	5.00%

A tolerance of 1 per cent. plus or minus will be allowed in the above. Impurities of over .25 per cent. will not be permitted.

NOTE: A high grade of composition metal, and an excellent bearing where speed and pressure are not excessive. Largely used for light castings, and possesses good machining qualities.

YELLOW BRASS

S. A. E. Specification No. 28

Copper	62.00 to 65.00%
Lead	2.00 to 4.00%
Zinc	36.00 to 31.00%

Total impurities in excess of .50 per cent. will not be permitted.

NOTE: This alloy represents a high grade of yellow brass; is tough and possesses good machining qualities. Its use is suggested in preference to

ordinary commercial yellow brass castings, which are, generally speaking, a miscellaneous assortment of mixtures, some of them containing considerable amounts of iron (from 1 to 3 per cent.). This is very undesirable, as it renders the castings liable to blowholes, hard spots and, in some cases, small particles of metallic iron.

CAST MANGANESE BRONZE

S. A. E. Specification No. 29

Manganese bronze is understood to mean a metal constituted principally of copper and zinc in the approximate proportion of 60 to 40, iron being present in small and manganese in variable quantities. Main dependence will be placed upon physical specifications.

Tensile Strength	60,000 lb. per sq. in.
Yield Point	30,000 lb. per sq. in.
Elongation in 2 inches	20%

NOTE: Manganese bronze is of value for castings where strength and toughness are required. Specifications are not severe, being easily met by all makers of quality castings. Test coupons should be attached to castings made in the sand, the use of chills, special sand or artificial methods of cooling being prohibited. This precaution prevents the use of inferior metals.

MANGANESE BRONZE SHEETS AND RODS

S. A. E. Specification No. 75B

Desired Composition:

Copper	56.00 to 60.00%
Tin	0.50 to 1.50%
Iron	0.50 to 1.50%
Manganese	0.10 to 0.75%
Lead and impurities, not over	0.25%
Zinc	Remainder

Manganese bronze is obtainable in cold-rolled sheets, strips and rods, cold-drawn rods, and hot-rolled sheets and rods.

Sheet manganese bronze in cold-rolled strips, sheets and rods, shall be furnished "annealed" or "hard-rolled." "Annealed" manganese bronze is to be designated as "light annealed," or "soft." "Hard-rolled" manganese bronze shall be furnished in the following tempers, and the amount of the reduction in thickness from the annealed sheet shall be as stated in the second table below.

Temper	Numbers Hard
Quarter hard	1
Half hard	2
Hard	4
Extra Hard	6
Spring	8

THICKNESS LIMITS

Thickness (B. & S. Gage) Limits:	Up to 5″ wide inclusive	Over 5″ wide to 8″ inclusive	Over 8″ wide to 11″ inclusive	Over 11″ wide to 14″ inclusive
No. 0000 to No. 0, Inc. (.4600″–.3248″)	.0044″	.0048″	.0051″	.0055″
Below 0 to No. 4, Inc. (.3248″–.2043″)	.0039″	.0043″	.0046″	.0050″
Below 4 to No. 8, Inc. (.2043″–.1254″)	.0034″	.0038″	.0041″	.0045″
Below 8 to No. 14, Inc. (.1254″–.0640″)	.0029″	.0033″	.0036″	.0040″
Below 14 to No. 18, Inc. (.0640″–.0403″)	.0025″	.0029″	.0033″	.0037″
Below 18 to No. 24, Inc. (.0403″–.0201″)	.0020″	.0024″	.0028″	.0032″
Below 24 to No. 28, Inc. (.0201″–.0126″)	.0016″	.0020″	.0024″	.0028″
Below 28 to No. 32, Inc. (.0126″–.0079″)	.0013″	.0017″	.0020″	.0024″
Below 32 to No. 35, Inc. (.0079″–.0056″)	.0010″	.0014″	.0017″	.0022″
Below 35 to No. 38, Inc. (.0056″–.0039″)	.0008″	.0012″	.0015″	.0019″.

NOTES FOR USE IN MAKING CALCULATIONS

Manganese bronze is one of the strongest of the non-ferrous alloys. It can be readily worked both hot and cold. It has a tensile strength of 70,000 to 72,000 lb., per square inch, with corresponding elongations of 30 to 25 per cent. in 2 inches, and an elastic limit of 30,000 to 35,000 lbs. Cold rolling or drawing increases the tensile strength and elastic limit and reduces the elongation correspondingly.

HARD CAST BRONZE

S. A. E. SPECIFICATION No. 43

Composition.—Copper 87 to 88%; tin 9.5 to 10.5%; zinc 1.5 to 2.5%.

NOTE: This is identical with U. S. Government Bronze "G;" having a tensile strength of approximately 35,000 lb. per square inch. A strong general utility bronze for severe working conditions where heavy pressures and high speeds obtain, for light gears, valves, etc.

GEAR BRONZE

S. A. E. SPECIFICATION No. 44

Composition.—Copper 88 to 89%; tin 11 to 12%; phosphorus 0.15 to 0.30%.

NOTE: This bronze is commonly known as English Gear Bronze and is used extensively in Europe and this country. It is very serviceable for gears and worms where the requirements are severe; especially when quiet running is a desired feature. Some makers temper this alloy with a ferrous hardener, using quantities up to 4 per cent., which gives excellent results.

ALUMINUM ALLOYS

No. 1

S. A. E. SPECIFICATION No. 30*

Aluminum, not less than........................ 90.00%
Copper.................................... 8.50 to 7.00%

Total impurities shall not exceed 1.7 per cent., of which not over 0.2% shall be zinc. No other impurities than carbon, silicon, iron, zinc and manganese shall be allowed.

NOTE: This is one of the lightest of the aluminum alloys, possessing a high degree of strength, and can be used where a tough, light alloy of these characteristics is required in automobile construction.

No. 2

S. A. E. Specification No. 31

Aluminum, not less than	80.00%
Zinc, not over	15.00%
Copper, between	2.00 and 3.00%
Manganese, not to exceed	0.40%

Total impurities shall not exceed 1.65 per cent., of which not more than 0.50 per cent. should be silicon, not more than 1.00 per cent. iron, and not more than 0.15 per cent. lead.

NOTE: The mixture possesses strength, closeness of grain, and can be cast solid and free from blowholes. It is a light metal, its specific gravity being in the neighborhood of 3.00.

No. 3

S. A. E. Specification No. 32

Aluminum	65.00%
Zinc	35.00%

Total impurities in excess of 1.65 per cent. will not be permitted.

NOTE: This is a mixture that can be used where cheap castings not to be subjected to any great strains are desired. It is a desirable mixture for flat plates, foot-boards, running boards, etc.

CHAPTER XXVIII

HARDENING STEEL[1]

1. Originally the name steel was applied to various combinations of iron and carbon, there being present, together with these, as impurities, small proportions of silicon and manganese. At the present time, however, the use of the name is extended to cover combinations of iron with tungsten, vanadium, nickel, chromium, molybdenum, titanium and some of the rarer elements. These latter combinations are quite generally known as the alloy steels to distinguish them from the carbon steels, in which latter the characteristic properties are dependent upon the presence of carbon alone. The alloy steels are divided into the high-speed steels and the Mushet or air-hardening steels. The specific properties that distinguish these different steels are due in part to their respective compositions, that is, to the particular elements they contain, and, in part, to their subsequent working and heat treatment.

EFFECT OF DIFFERENCE IN COMPOSITION OF STEEL

In general, any change in the composition of a steel results in some change in its properties. For example, the addition of certain metallic elements to a carbon steel causes, in the alloy steel thus formed, a change in position of the proper hardening temperature point. Tungsten or manganese tend to lower this point, boron and vanadium to raise it; the amount of the change is practically proportional to the amount of the element added. Just as a small proportion of carbon added to iron produces steel which has decidely different properties than those found in pure iron, so increasing the proportion of carbon in the steel thus formed, within certain limits, causes a variation in the degree in which these properties manifest themselves. For example, consider the property of tensile strength. In a "ten-point" carbon steel (one in which there is present but .1 per cent. of carbon) the

[1] From Machinery's Reference Book, No. 63, Published by the Industrial Press, New York City.

tensile strength is very nearly 25 per cent. greater than that of pure iron. Adding more carbon causes the tensile strength to rise, approximately, at the rate of 2.5 per cent. for each .01 per cent. of carbon added.

EFFECT OF HEAT TREATMENT

With a steel of a given composition, proper heat treatments may be applied which, of themselves, will first alter in form or degree some of its specific properties, or second, practically eliminate one or more of these, or third, add certain new ones. Physical properties of size, shape and ductility are examples of the first case; an example of the second case is found in the heating of steel beyond its hardening temperature, which takes away its magnetism, making it non-magnetic; and an example of the third is the fact that a greater degree of hardness may be added to steel by the process of hardening. In this connection it must be understood that, strictly speaking, hardness is a relative term and all steel has some hardness.

There are three general heat-treatment operations, so considered; forging, hardening and tempering. In all of these the object sought is to change in some manner the existing properties of the steel; in other words, to produce in it certain permanent conditions.

The controlling factor in all heat treatment is temperature. Whether the operation is forging, hardening or tempering, there is for any certain steel and particular use thereof a definite temperature point that alone gives the best results in working it. Insufficient temperatures do not produce the results sought. Excessive temperatures, either through ignorance of what the correct point is, or through inability to tell when it exists, cause "burned" steel; this is a common failing, resulting in great loss. Very slight variations from the proper temperature may do irreparable damage.

Due to temperature variation alone, carbon steel may be had in any of three conditions: first, in the unhardened or annealed state, when not heated to temperatures above 1350°F.; second, in the hardened state, by heating to temperatures between 1350 and 1500°F.; third, in a state softer than the second though harder than the first, when heated to temperatures which exceed 1500°F.

THE HARDENING PROCESS

The hardening of a carbon steel is the result of a change of internal structure which takes place in the steel when heated properly to a correct temperature. In the different carbon steels, this change, for practical purposes, is effective only in those in which the proportion of carbon is between .2 per cent. and 2 per cent. that is, between "twenty-point" and "two" carbon steels, respectively.

When heated, ordinary carbon steels begin to soften at about 390°F. and continue to soften throughout a range of 310°F. At the point 700°F. practically all of the hardness has disappeared. "Red hardness" in a steel is a property which enables it to remain hard at red heat. In a high-speed steel this property is of the first importance, 1020°F. being a minimum temperature at which softening may begin. This is some 630°F. above the point at which softening commences in ordinary carbon steels.

The process of hardening a steel is best carried out in a closed furnace. Of the many sources of energy capable of producing the required heat, electricity offers the most attractive advantages. The electric resistance furnace, as now built in a variety of sizes of either muffle or tube chamber types, has one fundamental point of superiority over all coal, coke, gas, or oil-heated furnaces. It is entirely free from all products of combustion, the heat being produced by electrical resistance. This is important. It does away with the chief cause of oxidation of the heated steel. Further, the temperature of the electric furnaces can be easily and accurately regulated to, and maintained uniform at, any desired point.

In the actual heating of a piece of steel, several requirements are essential to good hardening: first, that small projections or cutting edges are not heated more rapidly than is the body of the piece, that is, that all parts are heated at the same rate, and second, that all parts are heated to the same temperature. These conditions are facilitated by slow heating, especially when the heated piece is large. A uniform heat, as low in temperature as will give the required hardness produces the best product. Lack of uniformity in heating causes irregular grain and internal strains, and may even produce surface cracks. Any temperature above the "critical point" of steel tends to

open its grain—to make it coarse and to diminish its strength—though such a temperature may not be sufficient to lessen appreciably its hardness.

CRITICAL TEMPERATURES

The temperatures at which take place the previously mentioned internal changes in the structure of a steel are frequently spoken of as the "critical points." These are different in steels of different carbon contents. The higher the percentage of carbon present, the lower the temperature required to produce the internal change. In other words, the critical points of a high-carbon steel are lower than those of a low-carbon steel. In steels of the commonly used carbon contents, there are two of these critical temperatures, called the decalescence point and the recalescence point, respectively.

DECALESCENCE AND ITS RELATION TO HARDENING

Everyone interested in the hardening of steel will have noticed the increasing frequency with which reference is made to the decalescence and recalescence points of steel, in articles appearing in the technical press from time to time. It is only during the past few years that this pecularity in steel has come to the front, and there are still very many who do not possess even a rudimentary knowledge of the subject. The somewhat obscure references one usually sees in the treatises on hardening will not help the man in the hardening shop very much to a better understanding of the matter, and therefore an elementary explanation of the phenomenon will be welcome to many. It may be quoted that, as a matter of history, hardening has been done with more or less success, from the days of the famous Damascus swords up to only a comparatively short time ago, without anyone having discovered that steel possessed such a peculiarity as decalescence, but nevertheless its relation to hardening has aways existed, and its discovery paved the way for much scientific investigation into a subject that had been previously controlled by rule-of-thumb.

The "decalescence" and "recalescence" or "critical points" that bear relation to the hardening of steel, are simply evolutions that occur in the chemical composition of steel at certain tem-

peratures during both heating and cooling. Steel at normal temperatures carries its carbon, which is its chief hardening component, in a certain form—pearlite carbon to be more explicit—and if heated to a certain temperature a change occurs and the pearlite carbon becomes cementite or hardening carbon. Likewise, if allowed to cool slowly, the hardening carbon changes back again to pearlite. The points at which these evolutions occur are the decalescence and recalescence or critical points, and the effect of these molecular changes is to cause an increased absorption of heat on a rising temperature and an evolution of heat on a falling temperature. That is to say, during the heating of a piece of steel a halt occurs, and it continues to absorb heat without appreciably rising in temperature, at the decalescence point, although its immediate surroundings may be hotter than the steel. Likewise, steel cooling slowly, will, at a certain temperature, actually increase in temperature although its surroundings may be colder. This takes place at the recalescence point.

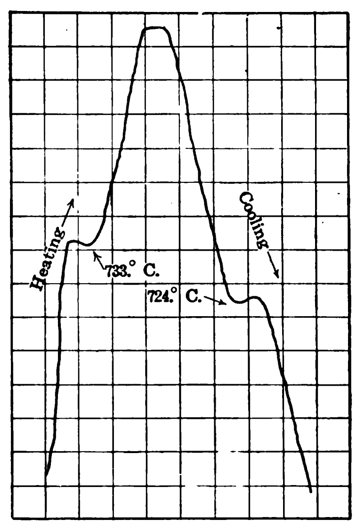

Fig. 128.—Decalescence and recalescence points in steel. Curve made by a recording pyrometer.

In Fig. 128 is shown a curve, taken on a recording pyrometer, in which the decalescence and recalescence points are well developed. From this it will be seen that the absorption of heat occurred at a point marked 733°C. (1350°F.) on the rising temperature, and the evolution of heat at 724°C. (1335°F.) on the falling temperature. The relation of these critical points to hardening is in the fact that unless a temperature sufficient to produce the first action is reached, so that the pearlite carbon will be changed to hardening carbon, and unless it is cooled with sufficient rapidity to practically eliminate the second action, no hardening can take place. The rate of cooling is material and accounts for the fact that large articles require to be quenched at higher temperatures than small ones.

A very important feature is that steel containing hardening

carbon, *i.e.*, steel above the temperature of decalescence, is non-magnetic. Anyone may demonstrate this for himself by heating a piece of steel to a bright red and testing it with an ordinary magnet. While bright red it will be found to have no attraction for the magnet, but at about a cherry-red it regains its magnetic properties. This feature has been taken advantage of as means of determining the correct hardening temperature, and appliances for its application are on the market. Its use is certainly to be recommended where no installation of pyrometers exists; the only point requiring judgment is the length of time an article should remain in the furnace after it has become non-magnetic. This varies with the weight and cooling surface, but may be tabulated according to weight, leaving very little to personal judgment.

RECAPITULATION

To sum up, the decalescence point of any steel marks the correct hardening temperature of that particular steel. It occurs while the temperature of the steel is rising. The piece is ready to be removed from the source of heat directly after it has been heated uniformly to this temperature, for then the structural change necessary to produce hardness has been completed. Heating the piece slightly more may be desirable for either or both of the two following reasons. First, in case the piece has been heated too quickly, that is, not uniformly, this excess temperature will assure the structural change being complete throughout the piece. Second, any slight loss of heat which may take place in transferring the piece from the furnace to the quenching bath may thus be allowed for, leaving the piece at the proper temperature when quenched.

If a piece of steel which has been heated above its decalescence point be allowed to cool slowly, it will pass through a structural change, the reverse of that which takes place on a rising temperature. The point at which this takes place is the recalescence point and is lower than the rising critical temperature by some 85 to 215 degrees. The location of these points is made evident by the fact that while passing through them the temperature of the steel remains stationary for an appreciable length of time. It is well to observe that the lower of these points does not manifest itself unless the higher one has been first fully passed. As these critical points are different for different steels, they cannot

be definitely known for any particular steel without an actual determination. While heating a piece of steel to its correct hardening temperature produces a change in its structure which makes possible an increase in its hardness, this condition is only temporary unless the piece is quenched.

QUENCHING

The quenching consists in plunging the heated steel into a bath, cooling it quickly. By this operation the structural change seems to be "trapped" and permanently set. Were it possible to make this cooling instantaneous and uniform throughout the piece, it would be perfectly and symmetrically hardened. This condition cannot, however, be realized, as the rate of cooling is affected both by the size and shape of the treated piece; the bulkier the piece, the larger the amount of heat that must be transferred to the surface and there dissipated through the cooling bath; the smaller the exposed surface in comparison with the bulk, the longer will be the time required for cooling. Remembering that the cooling should be as quickly accomplished as possible, the bath should be amply large to dissipate the heat rapidly and uniformly. Too small a quenching bath will cause much loss, due to the resulting irregular and slow cooling. To insure uniformly quenched products, the temperature of the bath should be kept constant, so that successive pieces immersed in it will be acted upon by the same quenching temperature. Running water is a satisfactory means of producing this condition.

The composition of the quenching bath may vary for different purposes, water, oil or brine being used. Greater hardness is obtained from quenching, at the same temperature, in salt brine and less in oil, than is obtained by quenching in water. This is due to a difference in the heat-dissipating power possessed by these substances. Quenching thin and complicated pieces in salt brine is unsafe as there is danger of the piece cracking, due to the extreme suddenness of cooling thus produced.

In actual shop work the steel to be hardened is generally of a variety of sizes, shapes and compositions. To obtain uniformity both of heating and of cooling, as well as the correct limiting temperature, the peculiarities of each piece must be given consideration in accordance with the points outlined above. In other words, to harden all pieces in a manner best adapted to but one

piece would result in inferior quality and possible loss of all except this one. Each different piece must be treated individually in a way calculated to bring out the best results from it.

THEORY OF CRITICAL POINTS

The presence of the critical points in the heating and cooling of a piece of steel is a phenomenon. The most reasonable explanation is as follows:

While heating, the steel uniformly absorbs heat. Up to the decalescence point all of the energy of this heat is exerted in raising the temperature of the piece. At this point, the heat taken on by the steel is expended, not in raising the temperature of the piece, but in work which produces the internal changes here taking place between the carbon and the iron. Hence, when the heat added is used in this manner, the temperature of the piece, having nothing to increase it, remains stationary, or, owing to surface radiation, may even fall slightly. After the change is complete, the added heat is again expended in raising the temperature of the piece, which increases proportionally.

When the piece has been heated above the decalescence point and allowed to cool slowly, the process is reversed. Heat is then radiated from the piece. Until the recalescence point is reached, the temperature falls uniformly. Here the internal relation of the carbon and iron is transformed to its original condition, the energy previously absorbed being converted into heat. This heat, set free in the steel, supplies for the moment, the equivalent of that being radiated from the surface, and the temperature of the piece ceases falling and remains stationary. Should the heat resulting from the internal changes be greater than that of surface radiation, the resulting temperature of the piece will not only cease falling but will obviously rise slightly at this point. In either event the condition exists only momentarily, but when the carbon and iron constituents have resumed their original relation, the internal heating ceases, and the temperature of the piece falls steadily, due to surface radiation.

RESULTS OBTAINED FROM SAMPLE SPECIMENS

In order to show graphically the necessity of working carbon steels at the proper temperature points, a series of specimen pieces

of the same steel were treated at different temperatures. The steel used contained exactly 1 per cent. carbon. A number of test specimens were made of this from adjacent parts of the same bar.

First the critical points of this steel were determined. Temperatures were recorded throughout both the heating and cooling. In the diagram, Fig. 129, these values have been plotted. The curve shows graphically the location of the critical points, and also the slight fall or rise of temperature as the case may be. With this data obtained, seven specimens of the same steel were heated, in the electric furnace, each to a different tempera-

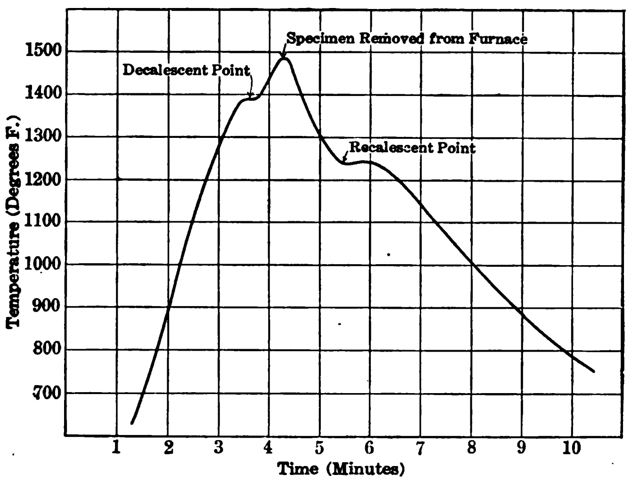

FIG. 129.—Diagram showing the relation between time and temperature when heating steel, and the critical temperatures of 1 per cent. carbon steel.

ture. As these pieces were removed from the furnace they were immediately quenched in water. The temperature of the quenching bath was held constant at 45°F. The hardened pieces were then broken at right angles and the fractured surface of each was photographed under a microscope. An inspection of the photographs at once showed the serious effects of overheating on the structure of the steel and hence on its strength.

One specimen was hardened just as the temperature reached the decalescence point. This showed clearly the direction in which the hardening moves, namely, from the exterior toward the interior. This would naturally be expected as the temperature of the surface, which is exposed directly to the source

of heat, reaches the critical point first. This condition indicates the necessity of heating the piece uniformly.

CONCLUSIONS

The hardening of carbon steels for highest quality and greatest saving entails, then, three things. First, a definite knowledge of what constitutes the correct temperature at which to harden the steel. The second point necessitates a positive means of accurately determining this hardening temperature for any carbon steel. The third consideration is that the correct hardening temperature, once determined, is actually carried out in the hardening work. A simple and effective way of doing this is by checking the temperature of the hardening furnace by means of a pyrometer. When there is a large quantity of work to be hardened, economy dictates a permanent installation of pyrometers. The convenience of such installations is manifest. A thermo-couple is placed in each furnace. A number of these, from three to sixteen, depending upon individual conditions, are connected by wire leads, through a selective switch to one meter. By a turn of the switch, the temperature of any furnace may be read at once from the meter. This makes it possible for the foreman to know definitely, at a single point, of all of the hardening furnaces in use.

HEAT TREATMENT OF ALLOY STEEL

The rapid development of the automobile industry in America has awakened a quick, keen appreciation of the great importance of proper heat treatment of steel. Scientific heat treatment is quite as essential as the quality of steel. Ordinary steel may acquire good physical qualities with proper heat treatment, and the best of steel can be ruined by defective methods. There must be thoroughness in the various operations of annealing, hardening and tempering, for only treatment carried on with care, makes uniformity of product possible. This is particularly true in the production of drop forgings.

The difference between ordinary and the best steel is great. For example, the elastic limit of ordinary steel is about 40,000 lb. per square inch, with a reduction of area of, say, 50 per cent. Nickel steel properly heat treated has an elastic limit

of 80,000 to 100,000 lb. per square inch of section, with a reduction of area of 50 per cent., or more. Brittleness does not follow proper heat treatment, the enduring quality being increased in a greater ratio than the elastic limit. Consequently, "fatigue," or whatever we name the cause of breakage, is less likely to develop in a properly heat-treated and tempered material than in an annealed and soft material. This fact, discovered in the laboratory and established in actual practice, is now commonly accepted by metallurgical experts, notwithstanding that it completely overturns previous general belief.

Another commonly accepted belief disproved is that strength and stiffness are coordinate, or "the stronger a piece of steel, the stiffer it is." To illustrate, it was thought if one piece of steel were twice as strong as another, it would bend only one-half as much under a given weight; but actual test has shown that a chrome-nickel steel having an elastic limit of 150,000 lb. or more per square inch of section, bends under a given load the same amount as a carbon steel specimen, and this condition holds true as long as the load is within the elastic limit of the weaker material. The elastic limit of a well-tempered steel spring is about 150,000 lb. per square inch, but a spring can be made of soft steel. If it is not loaded beyond its elastic limit, the spring will return to its original shape after every deflection, but the deflection would not be sufficient to make a good spring. In fact, it would be hardly noticeable and, of course, would be of little value. Between these extremes lie the steels used by the spring makers in the past.

Not only has the automobile industry forced the spring makers to depart from their old materials and methods, but the change extends all along the line. Assume that a .20 carbon steel has been used with advantage for a given design of crank shaft, neither bending nor breaking through a long continued use, and that the bearing surfaces are as small in area as can be used without heating or excessive wear. A crank shaft of properly treated chrome-nickel steel, having an elastic limit four or five times as high as the .20 carbon steel would be no stiffer, but would have greatly increased life and reliability.

NICKEL STEEL

Nickel steel is the most generally used of the alloy steels. The best quality contains .20 to .25 per cent. carbon, 3.50 per cent.

nickel, .60 to .90 per cent. manganese, and not over .04 per cent. sulphur and phosphorus. A slightly lower carbon content is sometimes used for case-hardening purposes, and a higher carbon percentage is much used for crank shafts. Nickel steel is usually made in the basic open-hearth furnace. It is an excellent steel for case-hardening, and is easier to machine than other alloy steels.

Chrome-vanadium steel made with high carbon content is suitable for oil-hardened gears and springs. When made with a low carbon content it is used for case-hardening gears, and, when oil quenched and annealed, for axles, shafts and steering knuckles. When a better material than the best nickel steel is needed, the various kinds of chrome-vanadium steel are to be recommended. They can be easily forged and can be machined more readily than chrome-nickel steels of corresponding carbon percentages.

CHROME-NICKEL STEELS

Chrome-nickel steels are made either with a high carbon content, and used for oil-hardened gears and springs, or with a low carbon content, in which case the steel is used for axles, shafts, forged parts, and case-hardened gears. The high carbon steel carries about .5 per cent. of carbon, while the low carbon alloy carries .25 per cent. The nickel content is from 2 to 3.5 per cent., while the chromium varies from 1 to 1.5 per cent. A special nickel-chrome-tungsten steel is sometimes used for springs. Nickel-chrome steels possess excellent static qualities, but present difficulties in heat treatment, forging and machining.

Silico-manganese and silico-chrome steels with medium and low carbon contents are used to a considerable extent abroad for springs and gears. Their relatively low cost favors their use, but they do not stand up well when subjected to shocks, and are too sensitive to heat treatment. When handled with great care they give good results where the temperature for the heat treatment can be accurately gaged. Chrome steels with high carbon content are used to a considerable extent for balls and ball races. Tungsten steels are universally used for making magneto magnets.

IMPORTANCE OF HEAT TREATMENT OF ALLOY STEELS

While the best alloy steels are none too good for most of the parts in automobile construction, their qualities will not become

pronounced unless they receive proper heat treatment. It is waste of money to buy good alloy steels without knowing how to properly treat them to bring forth their exceptional qualities. For gauging the heat a pyrometer is necessary, but it is too often supposed to take care of itself.

The heat treatment operations depend upon established scientific facts, and a lack of appreciation of this causes many people to buy high-priced alloy steels from which they get no better results than from carbon steel properly handled. As an example of the effect of heat treatment may be mentioned a chrome steel which in its rolled condition had an elastic limit of 158,000 lb., 5 per cent. elongation, and 9.4 per cent. reduction in area. The same steel, oil tempered and annealed, had an elastic limit of 153,000 lb., 14 per cent. elongation and 52 per cent. reduction in area. In other words, the material was transformed from brittle to tough without appreciably affecting its elastic limit. Nickel steel similarly treated will have the elastic limit raised 20 per cent., with its elongation unchanged and its reduction in area improved.

CASE-HARDENED VS. OIL-HARDENED GEARS

Both case-hardened and oil-hardened gears are largely used in automobile construction. As previously mentioned, the chrome-vanadium, chrome-nickel and silico-manganese alloys are made with both high and low carbon contents. The former contains about .45 to .60 per cent. carbon and enough other hardening elements so that by merely quenching the steel in oil from a bright red heat, surface hardening is produced sufficient for ordinary wearing purposes, while the hardness does not penetrate deeply into the gear, but leaves a tough and strong core. The low carbon alloy steels, with about .20 per cent. carbon, require to be case-hardened in order to produce sufficiently hard surface for wearing purposes. The author's observations lead him to prefer the case-hardened gear, the following conclusions being based on the results of direct tests on thousand of gears.

1. The static strength of case-hardened gears is equal to that of oil-hardened gears, assuming that in both cases steel of the same class of appropriate composition has been used, and the respective heat treatments have been equally well and properly conducted.

2. Direct experiments prove that case-hardened gears resist shocks better than oil-hardened.

3. The case-hardened gear resists wear incomparably better, although it is perhaps not as silent in action.

The strong objection to the case-hardening is in nine cases out of ten doubtless due to the fact that the case-hardening operations are not properly understood. The depth of the hard case or covering, the time and temperature required to produce certain results, and the exact control of the conditions, together with an accurate knowledge of the material to be treated, are factors which enter into successful case-hardening.

To obtain the best results in case-hardening ordinary carbon steel, the following rules should be observed. Steel containing less than .12 per cent. of carbon, and with a low percentage of manganese (less than .30 per cent-) should be used; the case-hardening should be accomplished by a chemically definite material, such as a mixture of 60 per cent. charcoal and 40 per cent. barium carbonate, and at a temperature between 1560 to 1920°F. The higher the temperature, the more rapid will be the the case-hardening. After the case-hardening operation, allow the steel to cool down to about 1100°F. Then re-heat the work to be case-hardened, and quench it at 1650°F. This heating and quenching has the effect of toughening the center, but the outside will be coarse-grained and brittle; therefore heat the material a second time to 1470°F. to render the outside non-brittle.

This procedure is more elaborate than that most commonly used, in which pieces are dumped directly from the case-hardening boxes into water. The process, however, can be somewhat modified if one uses a good grade of nickel steel, low in carbon, and after having case-hardened it at the appropriate temperature, permits the material to cool off in the boxes before re-heating and quenching. In this case, if the material is re-heated but once to 1470°F. the result will be fully equal to or better than those obtained by the most careful annealing and double quenching of ordinary carbon steels. It is, however, better to give a double quenching, as then extraordinary toughness and wearing qualities are obtained.

An ideal way of making a nickel steel gear consists in first annealing the blank, then rough machining it approximately to size, and then re-annealing before taking the last finishing cut. The gears are then packed in a mixture as mentioned, heated to a

temperature of about 1625 to 1650°F., carbonizing to a depth of about $\frac{1}{64}$ to $\frac{1}{32}$ inch. The gears are then permitted to cool in the boxes, are heated in 1500°F., and quenched in a hot brine or calcium-chloride solution, and finally re-heated to 1375 or 1400°F. and quenched in oil. The temper need not be drawn.

Another important point is that of drop forging small parts which can also be made from bars in automatic machines. No steel is improved by drop forging, although some steels are less susceptible to injury than others. In drop forging work, in order to give plasticity, the material must be heated very hot. An investigation of drop forging and bar cut gears, the former being the product of one of the foremost drop forging companies, showed that under static test the bar cut gears were fully 25 per cent. stronger and their resistance to shock was also greater.

CASE-HARDENING

The following contains an abstract of a paper read by Mr. David Flather before the Cycle Engineers' Institute, Birmingham, England.

The term "case-hardening" naturally implies the hardening of the skin of an article, and in order to fully understand the process and its object we must briefly consider the facts and laws upon which it is founded. Carbon has a very great affinity for iron and combines with it at all temperatures above faint red heat. Advantage is taken of this fact in the production of steel by cementation followed by water or oil-hardening.

For many purposes in machine work we require articles to have a perfectly hard surface and yet be of such a nature that there is no chance of their breaking in use. In many instances this result can be obtained with high-class crucible steel, but for axles, cups, cones, and many similar parts, it is extremely difficult to obtain perfect hardness combined with great resistance to torsional, shearing, or bursting strains. For such purposes nothing can meet these requirements so fully as articles which have been case-hardened. The greatest risks in the employment of all steel often occur during its treatment by the producer, and whether it be the finest cast steel or only common Bessemer, it is of first importance that it should be carefully and properly treated with a view to the work it has to do.

Both iron and mild steel have been employed as material for case-hardening; but this is the "steel age," and iron has long passed its day. The steel employed should be prepared, selected, and controlled from the beginning with the object of suiting it to its requirements. There are, of course, many points relating to its composition and treatment by the producer which can only be gained by long experience and by study of the requirements. Suffice it to say that the steel used should be low in carbon and capable of absorbing more carbon with great uniformity when heated under proper conditions; it should contain a minimum of deleterious impurities, and be perfectly sound and free from mechanical faults or weaknesses caused by over-heating during the manufacturing processes.

The carbonizers in general use at the present day are animal charcoal, bones, and one or two other compositions sold under various names, consisting of mixtures of carbonaceous matter and certain cyanides or nitrates. For very slight hardening, cyanides alone are still found very useful, but no great depth of casing is ever attempted with these. Theoretically, the perfect carbonizer should be a simple and pure form of carbon, and good charred leather gives the most certain and satisfactory results. Care should be taken to avoid poorly charred leather or that made from old boots, belting, etc.

FURNACE HEAT

The proper heat for case-hardening is about 1800°F., or a full orange heat and this should be maintained with great regularity throughout the operation. The length of time occupied in carbonizing is regulated by the depth of casing required, and indirectly by the dimensions of the article. At the close of the carbonizing period the pot is withdrawn from the furnace and placed in a dry place, where it is allowed to become quite cold. It is then opened, the articles taken out and brushed over to remove all adhering matter. If the pot has been properly packed and luted up, the articles should be quite white, or at least have only a slight film or bloom of a deep blue color; the denser and more inclined to redness is the surface, the more imperfect has been the packing and sealing of the pot.

RE-HEATING AND HARDENING

The carbonized articles are now placed in a muffle furnace and steadily raised to a good cherry red (1470°F.), and then quenched in cold or tepid water or oil, according to the purpose of the articles required. They should remain in the cooling liquid until they are quite cold right through the body of the metal; thus completing the process.

Although the proper temperature for case-hardening is about 1830°F., this temperature may be modified to suit the purpose in view. The absorption of the carbon commences when the steel reaches a low cherry-red heat (1300°F.); it begins, of course, at the outer surface and gradually spreads until the whole of the steel is carbonized. The length of time this requires depends upon the thickness of the metal being treated. The percentage of carbon absorbed is governed by the temperature, and although the increase of carbon is not in uniform proportion to the rising temperature throughout, it is perhaps sufficient for our present purpose to note that at 1300°F., iron, if completely saturated, can contain no more than about .50 per cent carbon; at 1650°F., about 1.5 per cent. carbon; and at 2000°F., about 2.5 per cent. These results, however, are only obtainable when the whole section of the iron has received all the carbon it is capable of absorbing at the given temperature, and is therefore in a state of equilibrium. From this it will be seen that if the process is stopped before the action is complete, the central parts of the iron must contain less carbon than the outside, and upon this fact the process of case-hardening is founded.

If we take two pieces of 5/8 inch diameter round mild steel, and heat one of them with a carbonizer at a cherry-red heat, and the other at a bright orange heat, for six hours, the first will be cased to a depth of about 1/32 inch, and the other to a depth of nearly 1/16 inch, while the amount of carbon taken up will be about .50 and .80 per cent. respectively; so that, so far as regards the hardness of the skin, the piece carbonized at the higher temperature gives the best result. From this we learn that a temperature of 1830°F., will give us sufficient hardness of case.

We have next to find which temperature has the least harmful effect on the mild steel core, and this can best be found by heating

pieces of the mild steel at varying temperatures at and above the selected one for the same length of time, using lime or other inert substance in the pot instead of a carbonizing material, and afterward reheating and quenching in water. Suppose, for example, we take three pieces, heating at 1830, 2370 and 2730° F., or full orange, white and bright white respectively. We shall find that those at 2370 and 2730° break very short and have lost nearly all their original tenacity, while that at 1830° appears tougher and altogether stronger than before.

Having arrived at a knowledge of the right temperature, it remains now to inquire as to the length of time requisite to yield a sufficient depth of case. At a full orange heat a bracket cup of ordinary dimensions should in 2 hours be hardened $1/32$ inch deep, and a bracket axle $11/16$ inch diameter in 6 hours would have a case $1/16$ inch deep. From this it will be seen that the speed of penetration is not in exact proportion to the time of heating.

RESULTS OF HARDENING WITHOUT REHEATING

We now arrive at that part of the process where a most important improvement has been made, *i.e.*, the final hardening by quenching in water. It formerly was customary at the end of the carbonizing period to open the pot and fling the contents headlong into a tank of cold water. Here and there some of the more careful workers took each article separately, but direct from the pot, and plunged it into water. These latter obtained better results, but even they had a great deal of trouble in the way of breakages and want of regular hardness. Finding that axles taken singly from the pot and quenched were better than those quenched in bulk,and that if allowed to cool down to cherry red they were better still, an application of the old rule to harden on a rising heat led to the now established principle of allowing the pot and its contents to become quite cold, afterward reheating to cherry red and quenching with water. By this means we obtain a case of great hardness with a very tough core—that is, of course, provided a suitable steel is employed.

To understand the reason of this improved method of working we must remember that the exterior of the steel is now about .80 per cent. carbon, and that steel of all kinds raised to and maintained at the high temperature employed for case-hardening will, unless subjected to mechanical work, show evidence of over-

heating, being very brittle and liable to easy fracture; the metal has little or no cohesion and readily wears away. Steel so hardened breaks with a very coarse crystalline fracture, in which the limits of the case are badly defined. It is known that when steel is gradually heated there is a certain point at which a great molecular change takes place, and that perfect hardness can only be obtained by quenching at this critical point. If quenching takes place below the critical temperature, the steel is not sufficiently hard; if above, though full hardness may be obtained, strength and tenacity are lost in part or completely, according as the critical temperature is exceeded by much or by little. This critical point lies beween 1380 and 1470°F., or cherry-red color heat. It may be asked why it is not sufficient, when taking the article out of the pot, to allow it to cool down to cherry red and then quench it. To this the answer is that the high temperature has already created a coarsely crystalline condition in the steel, and that until it has become quite cold and has again been heated up to the critical temperature, a suitable molecular condition cannot be obtained. When steel is cooled, whether slowly or not, it bears in its structure a condition representative of the highest heat it was last subjected to.

CASE-HARDENING PRACTICE OF PENNSYLVANIA RAILROAD SHOPS

It may be of interest to note the case-hardening practice followed by the Altoona shops of the Pennsylvania Railroad Company. The compound for case-hardening is made from 11 lb. prussiate of potash, 30 lb. sal soda, 20 lb. coarse salt, and 6 bushels powdered charcoal (hickory preferred). These ingredients are mixed thoroughly, using 30 quarts of water in mixing. The following method is pursued in packing the material to be case-hardened. The bottom of the box is covered to a depth of 2 inches with the compound. The parts to be hardened are to be placed solidly so that the compound is in contact with the bottom surface of the part, care being taken that the work does not touch the sides of the box or other pieces. After the first layer of the material is placed, it is covered on all sides and on top with the compound and solidly packed. After the first course is packed the process is repeated, care being taken to have a sufficient amount of compound between every course.

There should not be less than 2 inches of compound on the top of the last course. Then the lid is thoroughly sealed with a luting of fire clay.

In the furnace the box rests on rollers to allow the flames to pass under it. When the material has been in the furnace a sufficient length of time, the box is withdrawn to a trestle flush with the floor of the furnace and parallel with and close to a water tank, after which the material is removed from the box and plunged into the water. This method makes it possible to obtain a depth of case on large material of from 1/16 to 5/32 inch in 14 hours, and of about 1/16 inch on bushings and small parts in from 2½ to 3 hours. All parts to be case-hardened are thoroughly cleaned so as to be free from oil or grease.

CHAPTER XXIX

METHODS OF TESTING THE HARDNESS OF METALS[1]

Few properties of iron and steel are of more importance than that of hardness. In some cases, as with a cutting tool or a pressure die, the metal is practically valueless unless it can retain a sharp edge; while in other instances, where the material has to be machined or cut or trued to shape, even a relatively slight increase of hardness is the cause of much inconvenience and expense. In a third class of material a good wearing surface is of prime importance; while, lastly, hardness may often serve as an indication of a degree of brittleness and untrustworthiness which might perhaps be otherwise unsuspected.

Hardness may be defined as the property of resisting penetration, and, conversely, a hard body is one which, under suitable conditions, readily penetrates a softer material. There are, however, in metals various kinds or manifestations of hardness according to the form of stress to which the metal may be subjected. These include chiefly tensile hardness, cutting hardness, abrasion hardness, and elastic hardness.

Comparison will be made in the following of four typical methods of measuring hardness. Those selected include the sclerometer introduced by Professor Thomas Turner about 1886; the scleroscope invented by Shore; the form of indentation test adopted by Brinell about 1900; and the drill test introduced by Keep a few years earlier. Each of these methods has been used in actual works practice, and may thus be regarded as being typical of the particular class of test to which it belongs. Among the many other forms of test the microsclerometer and wearing tests call for special mention, though to these only incidental reference can be made.

The principles underlying the four methods selected for comparison may be briefly described as follows:

[1] From Machinery's Reference Book, No. 62, published by The Industrial Press, New York.

Turner's Sclerometer.—In this form of test a weighted diamond point is drawn, once forward and once backward, over the smooth surface of the material to be tested. The hardness number is the weight in grammes required to produce a standard scratch. The scratch selected is one which is just visible to the naked eye as a dark line on a bright reflecting surface. It is also the scratch which can just be felt with the edge of a quill when the latter is drawn over the smooth surface at right angles to a series of such scratches produced by regularly increasing weights.

Shore's Scleroscope.—In this instrument, a small cylinder of steel, with a hardened point, is allowed to fall upon the smooth surface of the metal to be tested, and the height of the rebound of the hammer is taken as the measure of hardness. The hammer weighs about 40 grains, the height of the rebound of hardened steel is in the neighborhood of 100 on the scale, or about 6¼ inches, while the total fall is about 10 inches or 255 mm.

Brinell's Test.—In this method, a hardened steel ball is pressed into the smooth surface of the metal so as to make an indentation of a size such as can be conveniently measured under the microscope. The spherical area of the indentation being calculated, and the pressure being known, the stress per unit of area when the ball comes to rest is calculated, and the hardness number obtained. Within certain limits the value obtained is independent of the size of the ball, and of the amount of pressure.

Keep's Test.—In this form of apparatus a standard steel drill is caused to make a definite number of revolutions while it is pressed with standard force against the specimen to be tested. The hardness is automatically recorded on a diagram on which a dead soft material gives a horizontal line, while a material as hard as the drill itself gives a vertical line, intermediate hardness being represented by the corresponding angle between 0 and 90°.

COMPARISON BETWEEN TESTING METHODS

Each form of test has its advantages and its limitations. The sclerometer is cheap, portable, and easily applied, but it is not applicable to materials which do not possess a fairly smooth reflecting surface, and the standard scratch is only definitely recognized after some experience. The Shore (scleroscope) test is simple, rapid, and definite for materials for which it is

suited, but further information is yet needed as to the exact property which is measured by this form of test. As shown by De Freminville, the result obtained varies somewhat with the size and thickness of the sample, while if the test-piece is supported on a soft material, such as a plasticine, the results are valueless. It should also be pointed out that india-rubber gives a rebound of 23, which is equal to that of mild steel, while light soft pine wood gives a rebound of 40, which is nearly twice as great as that of gray cast iron. Curiously enough, hard wood, like teak, gives a rebound of about 12, while some samples are considerably lower than this. As illustrating the influence of the support, a sample of exceptionally hard rolled copper, about .040 inch in thickness, when supported on a block of hard steel, and tested with the blunt or "magnified" hammer supplied, gave a value of 30, which was increased to 34 when the copper was supported on wood. A sample of brass only gave a value of 17, and yet this brass would scratch the copper, while the copper would not scratch the brass. From these results it would seem that the Shore test is only applicable to a certain class of substances. It appears to test what may be termed the "elastic hardness," and gives high results with metals in the "worked hard" condition. Tests appear to show that good results are, however, obtained with glass and with porcelain, as well, of course, as with most metals.

The Brinell test is specially useful for constructive material; it is easily applied and definite, and is now of all hardness tests the one most employed. It appears to give satisfactory results with wood, but cannot be applied to very brittle materials, such as glass, or to hard minerals. Keep's test is specially suited for castings of all kinds, as it records not merely the surface hardness, but also that of the whole thickness, and gives indications of blowholes, hard streaks, and spongy places. Obviously, it can only be applied to materials the hardness of which is less than that of hardened steel.

THE BRINELL METHOD OF TESTING THE HARDNESS OF METALS

The method of testing the hardness of metals devised by Mr. J. A. Brinell of Sweden, first made public in 1900 his method of testing the hardness of iron and steel. In working out his method, Brinell kept in view the necessity of taking into account

the requirements that the method must be trustworthy, must be easy to learn and apply, and capable of being used on almost any piece of metal, and particularly, to be used on metal without in any way being destructive to the sample.

Principle of Brinell Method.—The Brinell method, as mentioned, consists in partly forcing a hardened steel ball into the sample to be tested so as to effect a slight spherical impression, the dimensions of which will then serve as a basis for ascertaining the hardness of the metal. The diameter of the impression is measured, and the spherical area of the concavity calculated. On dividing the amount of pressure required in kilogrammes for effecting the impression, by the area of the impression in square millimeters, an expression for the hardness of the material tested is obtained, this expression or number being called the hardness numeral. In order to render the results thus obtained by different tests directly comparable with one another, there has been adopted a common standard with regard to the size of ball as well as to the amount of loading. The standard diameter of the ball is 10 millimeters (0.3937 inch) and the pressure 3000 kilogrammes (6614 pounds) in the case of iron and steel, while in the case of softer metals a pressure of 500 kilogrammes (1102 pounds) is used. Any variation either in the size of the ball or the amount of loading will be apt to occasion more or less confusion without there being any advantage to compensate for such inconvenience. Besides, making any comparisons between results thus obtained in a different manner would be more or less troublesome, and complicated calculations would be required.

The diameter of the impression is measured by means of a microscope of suitable construction, and the hardness numeral may be obtained without calculation, directly from a table worked out for the standard diameter of ball and pressures mentioned.

Relation between Hardness of Materials and Ultimate Strength.—It has been pointed out by Mr. Brinell that this method of testing hardness of metals offers a most ready and convenient means of ascertaining within close limits the ultimate strength of iron and steel. This, in fact, is one of the most interesting and important results of this method of measuring hardness. In order to determine the ultimate strength of iron and steel, it is only necessary to establish a constant coefficient

determined by experiments which serves as a factor by which the hardness numerals are multiplied, the product being the ultimate strength. Comprehensive experiments were undertaken with a considerable number of specimens of annealed material obtained from various steel works, for the purpose of establishing the coefficiency. The results obtained were as follows:

For hardness numerals below 175, when the impression is effected transversely to the rolling direction, the coefficient equals .362; when the impression is effected in the rolling direction, the coefficient equals .354.

For hardness numerals above 175, when the impression is effected transversely to the rolling direction, the coefficient equals .344; when the impression is effected in the rolling direction, the coefficient equals .324.

If the hardness numerals are multiplied by these coefficients, the result obtained will be the ultimate tensile strength of the material in kilogrammes per square millimeter. It is evident that coefficients can easily be worked out so that if the hardness numerals be multiplied by these the strength could be obtained in pounds per square inch.

Suppose, for instance, that a test of annealed steel by means of the Brinell ball test gave an impression of a diameter of 4.6 millimeters. Then the hardness numeral, according to our table, would be 170, and the ultimate tensile strength consequently $.362 \times 170 = 61.5$ kilogrammes per square millimeter, provided the impression was effected transversely to the rolling direction.

One of the greatest advantages of the Brinell method is that in the case of a large number of objects being required to be tested, each one of the objects can be tested without demolition, and without the trouble of preparing test bars.

APPLICATION OF THE BRINELL BALL TEST METHOD

Summarizing what has been said in the previous discussion, and adding some other important points, we may state the various uses for which the Brinell ball test method may be applied, outside of the direct test of the hardness of construction materials and the calculation from this test of the ultimate strength of the materials, as follows:

TABLE XXXIX.—HARDNESS NUMERALS

Steel ball of 10 millimeters diameter

Diameter of impression, mm.	Hardness numeral, pressure, kg.		Diameter of impression, mm.	Hardness numeral, pressure, kg.		Diameter of impression, mm.	Hardness numeral, pressure, kg.		Diameter of impression, mm.	Hardness numeral, pressure, kg.		Diameter of impression, mm.	Hardness numeral, pressure, kg.	
	3000	500		3000	500		3000	500		3000	500		3000	500
2.00	946	158	3.00	418	70	4.00	228	38	5.00	143	23.8	6.00	95	15.9
2.05	898	150	3.05	402	67	4.05	223	37	5.05	140	23.3	6.05	94	15.6
2.10	857	143	3.10	387	65	4.10	217	36	5.10	137	22.8	6.10	92	15.3
2.15	817	136	3.15	375	63	4.15	212	35	5.15	134	22.3	6.15	90	15.1
2.20	782	130	3.20	364	61	4.20	207	34.5	5.20	131	21.8	6.20	89	14.8
2.25	744	124	3.25	351	59	4.25	202	33.6	5.25	128	21.5	6.25	87	14.5
2.30	713	119	3.30	340	57	4.30	196	32.6	5.30	126	21	6.30	86	14.3
2.35	683	114	3.35	332	55	4.35	192	32	5.35	124	20.6	6.35	84	14
2.40	652	109	3.40	321	54	4.40	187	31.2	5.40	121	20.1	6.40	82	13.8
2.45	627	105	3.45	311	52	4.45	183	30.4	5.45	118	19.7	6.45	81	13.5
2.50	600	100	3.50	302	50	4.50	179	29.7	5.50	116	19.3	6.50	80	13.3
2.55	578	96	3.55	293	49	4.55	174	29.1	5.55	114	19	6.55	79	13.1
2.60	555	93	3.60	286	48	4.60	170	28.4	5.60	112	18.6	6.60	77	12.8
2.65	532	89	3.65	277	46	4.65	166	27.8	5.65	109	18.2	6.65	76	12.6
2.70	512	86	3.70	269	45	4.70	163	27.2	5.70	107	17.8	6.70	74	12.4
2.75	495	83	3.75	262	44	4.75	159	26.5	5.75	105	17.5	6.75	73	12.2
2.80	477	80	3.80	255	43	4.80	156	25.9	5.80	103	17.2	6.80	71.5	11.9
2.85	460	77	3.85	248	41	4.85	153	25.4	5.85	101	16.9	6.85	70	11.7
2.90	444	74	3.90	241	40	4.90	149	24.9	5.90	99	16.6	6.90	69	11.5
2.95	430	73	3.95	235	39	4.95	146	24.4	5.95	97	16.2	6.95	68	11.3

1. Determining the carbon content in iron and steel.
2. Examining various manufactured goods and objects, such as rails, tires, projectiles, armor plates, guns, gun barrels, structural materials, etc., without damage to the object tested.
3. Ascertaining the quality of the material in finished pieces and fragments of machinery even in such cases when no specimen bars are obtainable for undertaking ordinary tensile tests.
4. Ascertaining the effects of annealing and hardening of steel.
5. Ascertaining the homogeneity of hardening in any manufactured articles of hardened steel.
6. Ascertaining the hardening power of various quenching liquids and the influence of temperature of such liquids on the hardening results.
7. Ascertaining the effect of cold working on various materials.

MACHINE FOR TESTING THE HARDNESS OF METAL BY THE BRINELL METHOD

A simple apparatus is shown in Figs. 130 and 131, which was devised by the Usines G. Derihon of Loucin, Belgium. This firm is an important manufacturer of high-grade drop forgings for automobile work, and it originally developed the machine for use in its own plant. As may be seen, the pressure is applied by a hand-operated screw, and the press is small enough to be perfectly portable, weighing only about 12 pounds.

The sectional view in Fig. 131 shows the action of the apparatus most clearly. The work is placed on the platen *F*, which rests in a spherical seat on top of adjusting screw *G*. By means of this self-adjusting seat, the work gets an even bearing and a direct pressure, even though its under surface may be quite out of true. The purpose of the adjusting screw *G* is, of course, to give a rapid adjustment for the thickness of the work. It will take in about 3 inches as shown. The thread of *G*, while of coarse pitch, still lies within the angle of repose, so that it is not disturbed when the pressure is applied by levers *M*.

A differential screw mechanism is used for applying the pressure. This mechanism consists of the double handle *M*, keyed to the sleeve *D*, which is threaded into the stationary nut *E*, and on to the ram *C*; this latter is kept from revolving by a stop *K*, which enters a slot cut in the flange, and is provided with a threaded chuck for holding the ball *B*. Sleeve *D* is, it will be

seen, the only revolving member of this differential screw. The thread on the inside has a lead of 8 millimeters, while that on the outside has a lead of 7.25 millimeters. This gives an advance of $8 - 7.25 = .75$ millimeters, or about .03 inch per revolution of the levers *M*. The advantage of this construction is, of course, that it gives the effect of a fine thread with a lead .03 inch, with the use of threads coarse enough to withstand the great strain to which they are subjected in tightening down the work.

FIG. 130.—Apparatus for testing the hardness of metal by the Brinell method.

The most ingenious feature of the mechanism is the provision made for gaging the pressure with which the ball is forced into the work. By the arrangement used, the frame *A* of the press itself serves as the spring by which the pressure is measured. As the ball is forced into the work with greater and greater pressure, the frame *A* is deflected and *B* and *F* spring apart. Arm *H* is screwed to *A* at its lower end as shown, but is free at its upper end. Here it is provided with a bearing point *X* close to fulcrum

Y of pointer P. As the frame A springs under the pressure, H, being free at the upper end, remains undistorted and stationary, while pivot Y rises. As pivot Y rises, lever P swings downward, since it rests only on the point X of stationary arm H. The lower end of lever P is provided with a thin metal disk R, which is thus swung in an arc of a circle about center Y. A series of holes Z, bored in the side of the frame, permit the position of this disk to be seen. These holes are so calibrated that each reads to a definite number of kilograms of pressure when disk R is

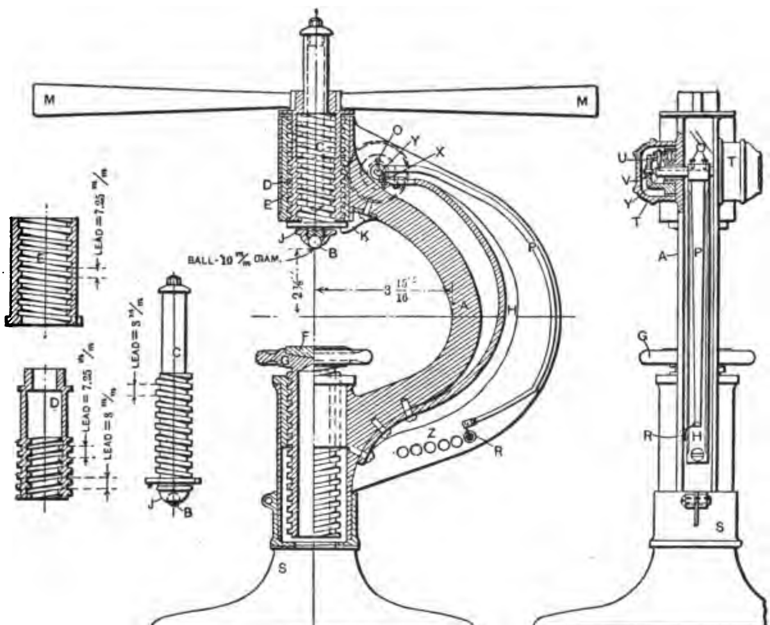

FIG. 131.—Details of apparatus shown in Fig. 130.

centered with it. Under the extreme tension when central with the left-hand hole, the reading shows the application of a pressure of 3000 kilograms or 6614 pounds.

The construction of the pivot joint at Y is interesting, and is best shown at the right of the engraving in Fig. D. The hub of pointer P is clamped to Y by two set-screws O, which set in a V-slot cut in Y. Two caps T are screwed on at either side to protect the bearings of pivot Y. Set-screws V are adjusted to take up the end motion of Y; they do not, however, restrain it in any direction other than the longitudinal. The real bearing is

furnished by the points of screws *U* (one at each end) in the bottoms of the V-grooves. These furnish a knife edge, or rather, point support, which gives the utmost freedom and sensitiveness of movement to the pointer *P* and the indicating disk *R*. The various adjustments in connection with this bearing, and the various contact points in the lever system, will be clearly understood from the engraving.

Figure 130 shows the instrument engaged in testing the hardness of a sprocket-wheel. The simplicity of its use will be immediately appreciated. The sprocket-wheel is placed on the platen, the adjusting screw is run up until the work makes contact with the ball, and then the handles are revolved until the indicating disk is centered with the particular hole in the frame, which shows that the standard pressure has been reached. Handles *M* are then screwed back again, the work is removed, and the diameter of the impression in millimeters measured. This gives the hardness number directly. The whole operation is evidently one of seconds only.

The S. A. E. gives the following rules in referring to hardness tests by the Brinell method:

The diameter of the ball shall be 10 mm. ± .0025 mm. The weight applied shall be 3000 kg. (6600 lb.) for 30 seconds.

The average diameter of the impression shall be obtained from two measurements at right angles to each other, made with an instrument having a reading error not over .05 mm.

The surface of the specimen shall be free from scratches. The specimen shall be taken deep enough to represent the true composition of the material to be tested, and shall be maintained in a plane normal to the direction of the testing load.

This test should not be used on soft steels less than ½ in. thick or on areas small enough to permit deflection of the edges of the specimen due to flow from the ball depression.

The hardness numerals are those given in the last table.

THE SHORE SCLEROSCOPE

The Shore scleroscope is an instrument in a measure dependent on sensitive touch; or, in other words, it feels the substance much the same as the human fingers. When we touch two or more objects, as; for instance, an orange and an apple, we know that the orange is softer because it yields under pressure more than

the apple. We are powerless to measure the hardness of any object that is harder than the finger tips, and there is no way of telling how hard it may be by finger pressure alone.

The sensitive touch of the scleroscope is produced by a tiny hammer dropping from a height of about 10 inches on to the metal, hardened steel, etc., which it penetrates slightly. The hammer moves freely, yet snugly, within a glass tube, and weighs about 40 grains. Its striking point consists of an inserted diamond of rare cleavage formation, annealed sufficiently to withstand shocks. This jeweled point is slightly convex and has an area of from about .010 to .025 square inch. When the plunger strikes the metal to be tested, it reacts or rebounds. The height of this rebound is read on a graduated scale, and an accurate determination of the quantitative hardness of the piece under test is thus obtained.

HARDNESS VS. ELASTICITY

When the hammer of the scleroscope is allowed to drop with no other force than its own weight, and the point is so flat that absolutely no impressions is made on the surface of very hard steel, then the rebound will be about 90 per cent. of the fall. This phenomenon is known as the elasticity of solid bodies. Now, since hardness is resistance to penetration, in its clearest definition, it stands to reason that the point of the hammer must be somewhat reduced and rounded. Therefore the relation between the weight of the hammer and its point should be such that when it drops on hardened steel, a permanent impression must always be made, so that if we had not the rebound to go by, the microscope would still show the values. When the area of the hammer is thus reduced enough to make a permanent impression, a certain amount of the energy stored in the hammer is utilized in doing work. This overcomes the tendency of the metal to resist penetration, depending on how hard it is, or the resistance it offers, and naturally it must rebound considerably less. The hammer always delivers a blow of exactly the same force. If now we get a rebound of 75 per cent. on very hard steel, we know that 15 per cent. of the hammer's energy was spent in its efforts to overcome the resistance of the steel before it had a chance to react and repel the missile.

THE INSTRUMENT

While the absolute weight of the entire hammer is little, it is very great relative to the striking area. The hammer has a cylindrical body and is guided in its fall by a glass tube. Great difficulty has been experienced in obtaining tubes with a sufficiently perfect bore. There seems to be no commercial method of manufacturing such tubes, and the method of "test-and-reject" is therefore employed, resulting in a very great amount of waste.

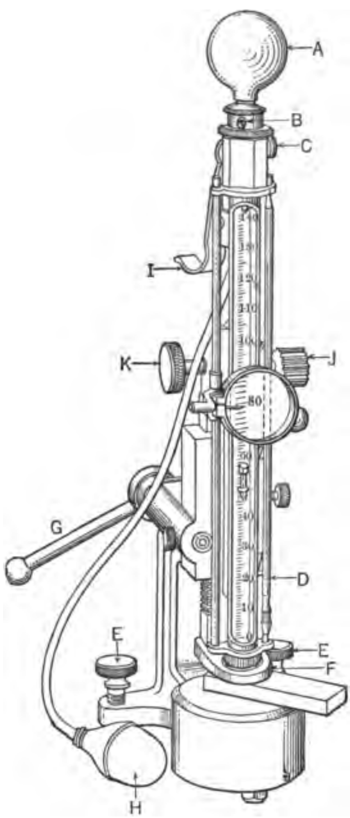

FIG. 132.—The Shore scleroscope for testing the hardness of metal.

The glass tube is secured to a frame in a vertical position with the lower end open. The operation of the instrument is very simple. When the hammer is to be raised to the top, the bulb *A*, Fig. 132, is pressed and then suddenly released. This sucks up the jeweled plunger hammer referred to so that it may be caught by a hook which is suspended exactly central in the glass tube and gauges with an internal groove on the top of the hammer. Adjusting screws *B* for the hook and its spring are contained in the removable knurled cap. *C* is a cylinder and piston for releasing the hook and hammer by bulb *H* whenever a test is to be made; *I* is a hook which is pressed at the same time and which opens a valve letting in the air and thus preventing the occurrence of a vacuum when the hammer drops. At *J* is shown a pinion knob for moving the instrument up and down independently of the heavy rack and clamp *F* actuated by the lever *G*. At *E* are shown leveling screws and at *D* a plum rod.

In using the instrument it is necessary that the actual point tested should be clean and horizontal and that the piece should

be firmly held. If necessary to test more than once, the piece should be slightly moved, so as to expose a fresh point to the hammer. The indentation made is, however, very minute, so that several are usually unobjectionable. When the ends of rods, drills, and many other tools are to be tested they are to be clamped in a bench vise and a swinging arm is employed

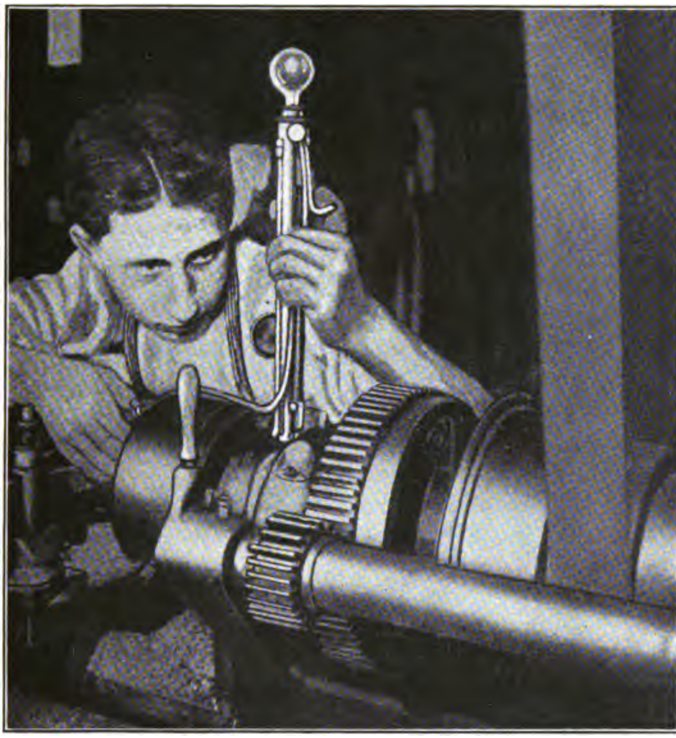

FIG. 133.—The scleroscope used to determine relative hardness of shaft and box of a lathe.

to hold the instrument. A kind of female dove-tailed finger ring attached to the clamp on the dove-tail rack bar of the instrument is provided for use in free-hand testing on very large floor work, on parts of machinery being assembled, or on the stock rack, etc. In Fig. 133, is shown how a shaft and box may be tested to determine the relative hardness. From what has been said it is apparent that the apparatus is of universal application.

Instead of dividing the whole length of the fall of the hammer into a scale consisting of 100 divisions, the figure 100 is carried

down to a point representing about 68 per cent. of the total height of the scale as shown in Fig. 132. This was not an arbitrary provision, but was adopted after consultation with leading metallurgists, one of whom was Dr. Paul Herault, of aluminum and electric steel making fame, of France. These authorities agreed that in the scleroscope, hardened steel of average hardness should be taken as the standard with which all other less hard metals should be compared; 100 is the average hardness of hardened steel; 90 is a low value, while 110 is a high value. The scale, therefore, makes it an easy matter to compare the various metals, no matter what their hardness is, and the rebound of the hammer is, therefore, measured against a scale graduated from 0 to 140. This scale is secured in position back of the glass tube. To aid in reading the rebound, a magnifying glass is supplied. After some practice the assistance of this glass may be dispensed with. However, when used, it is secured in such a position as to cover the probable region of the expected reading. The rod to the left of the tube is the support to which the magnifying attachment is secured and along which it is adjusted. The rod to the right of the tube is a plum rod; it swings freely from a point of attachment above, and enables the operator to keep the glass tube vertical.

With the scale graduated from 0 to 140, with hardened steel at or near 100, the hardness of all ordinary materials can be measured.

AMOUNT OF PRESSURE FOR INDENTATION

When the hammer falls through a height of 10 inches unto hardened steel, it will deliver a striking energy equal to about 20,000 times its own weight, acting through a very short space, or course. With a hammer weighing about 40 grains, and an indentation of, say, .002 inch depth, a working pressure of about 100 lb. is obtained. This force acting on a convex point about $\frac{1}{64}$ inch diameter, is concentrated. The pressure thus available is about 500,000 lb. per square inch, which is ample to exceed the elastic limit of the hardest and strongest steel in existence.

A remarkable feature of this instrument is that it is self-compensating with regard to the energy of the hammer blows on the softer metals. This is due to the yielding of the material and the comparatively slow stoppage of the hammer. In lead, for exam-

ple, a deep impression is made. This requires a great amount of energy, which is nearly all spent in doing work, and there is very little rebound afterward—about 3° as against 110 for the hardest steel. The constant pressure developed by the hammer is thus only 12 lb. instead of 100 or more for good hard steel, and, of course, the pressures for intermediate hardness as on brass and soft or tempered steel are always in proportion to the physical hardness of the brass or steel.

The Brinell test and scleroscope (Shore) test are the two referred to in the chapter on specifications of S. A. E. steels. Most manufacturers of motors use either one or both of these methods as seen by the frequent reference on the working drawings given in this volume. For this reason these two have been described at length.

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